AN EXPERIMENTAL INVESTIGATION OF STEAM - WATER FOG FLOW IN AN ANNULAR CHANNEL

A Thesis Submitted
In Partial Fulfilment of the Requirements
for the Degree of
MASTER OF TECHNOLOGY

By
MRITYUNJAI PRASAD SHARMA

to the
Department of Mechanical Engineering
Indian Institute of Technology, Kanpur

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CERTIFICATE

This is to certify that this work on "An Experimental Investigation of Steam - Water Fog Flow in an Annular Channel", has been carried out under my supervision and has not been submitted elsewhere for a degree.

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SYNOPSIS

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An experimental set up for the study of pressure - drop, void fraction distribution and heat transfer characteristics of steam - water fog - flow in an annular channel has been designed, fabricated and installed. Void - fractions have been measured using the Gamma - ray attenuation method. A review of the literature is given.

The annular test channel is made up of 1 inch O.D. and 1/2 inch O.D. stainless steel tubes of 1/16 inch wall thickness. Actual length of the test channel is 72 inches, but the test - length is 36 inches only. The ranges of parameters studies are : pressure 100 psia and 85 psia; mass velocities 2.05×10^5 lbs/hr. sq. ft. to 2.66×10^5 lbs/hr. sq. ft., quality of the flow changes from 0.40 to 0.80.

Tables and Curves showing the effect of quality on non-dimensional pressure drop parameter and void fraction are presented at different pressures and flow rates. Also, curves showing the effect of mass velocity on pressure gradient along the channel are presented.

The experimental data for pressure drop and void fraction are in good agreement with the predictions of the Martinelli - Nelson separated flow model correlation. It is observed that increase of mass velocity causes a decrease in the dimensionless pressure drop parameter and that the pressure drop per unit length along the channel is more at the lower pressures than at higher pressures. Void fraction data show dependence on pressure but not on mass velocity.

Suggestions for further experimental and theoretical research are presented.

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NOMENCLATURE

A	Cross - sectional area of flow, ft^2
D	Diameter of test section, ft.
D_e	Equivalent Diameter, ft.
e	$e = 2.718$
f	Friction factor, dimensionless.
G	Mass flow rate $lb/hr\text{-}ft^2$ (Mass Velocity).
g	Acceleration of gravity $ft/sec.^2$
g_c	Conversion Ratio $lb\text{-}ft/lb\text{-}hr^2$
h	Enthalpy, heat transfer coefficient.
I	Radiation intensity.
K	Bankoff's Parameter
l	Length in the direction of flow, ft.
P	Perimeter, ft.
p	Pressure, Psi
Δp	Pressure Drop, Psi
Q	Volumetric flow rate.
q	Heat flux $Btu/hr.ft^2$
R	Radius, ft.
S	Slip Ratio
T	Temperature
u	Velocity in axial direction ft/hr
V	Velocity ft/hr
v	Velocity in normal direction ft/hr, Specific Volume ft^3/lb.
W	Total mass flow rate lb/hr.

x Flow Quality

x_s Static quality

z Length coordinate in the direction of motion.

<u>Greek Letters</u>:

α Void fraction, dimensionless

θ Angle, degrees

μ Viscosity lb/hr.ft.

ρ Density lbm/ft^3

σ Surface tension lb/ft

τ Shear Stress lb/ft^2

ϕ_{LO} $\phi_{LO} = \left[(dp/dz)_{TPF} / (dp/dz)_{LO} \right]^{1/2}$

<u>Subscripts</u> :

1 Refers to Component 1, Liquid Phase

2 Refers to Component 2 , Gas Phase

f Refers to liquid phase

g Refers to gas phase

GPF Refers to only gas mass velocity in the two phases of the flow

LPF Refers to only liquid mass velocity in the two phase flow

TPF Refers to two - phase flow

LO Refers to the liquid flow with total (of the two phases) mass velocity.

m Refers to mixture (two - phase)

v Refers to vapor phase

w Refers to wall condition

CHAPTER I

INTRODUCTION

1.1A General:

A phase is one of the states of matter and can be a solid, a liquid or a gas. Multiphase Flow is the simultaneous flow of a mixture of several phases. Two-Phase Flow is the simplest case of multiphase flow — flow of only two phases.

The term two-component is used to describe flows in which the phases do not consist of the same chemical substance. Thus the flow of Air - Water mixture is a two-component two-phase flow.

In this investigation, we are concerned with single component (steam - water) two-phase flow which will be refrred as two-phase flow.

Flow of two-phase mixtures occurs in boilers, condensers, refrigerators, natural gas pipe lines, air-lift pumps etc. However, the recent researches into this field has acquired a new impetus because of the occurrence of this phenomena in Nuclear Power Reactors.

1.1B Flow Patterns and Fog-flow:

Depending upon the void-distribution and velocity distribution of the flow field (of two-phase flow) the morphological

arrangement of the phases (and/or components) changes to give rise different flow patterns. The number of characteristic flow - patterns and the names used for them vary somewhat from investigator to investigator. However, for vertical up flow, the most general types of patterns that can exist (in order of increasing quality from single phase liquid to single phase vapor flow) are (20) as follows:

(i) Bubbly flow: In this type of flow spherical and spherical cap bubbles, small in size compared with the tube diameter are distributed across the pipe with some amount of crowding towards the centre;

(ii) Plug or Slug Flow: Large, approximately axially symmetric gas bubbles that almost bridge the tube cause the flow to alternate between being mainly liquid and mainly gas.

(iii) Churn or froth flow: In this phase the gas is highly dispersed and interacting in a continuous liquid.

(iv) Semi-annular flow: Here, large slugs of gas have lost their axially symmetric characteristic. This is similar to plug flow, but the slug is now more of an interacting gas part. Generally the liquid contains gas and is similar to that described as churn flow.

(v) Annular flow : Annular flow consists of a gas core which usually contains liquid drops, with the main flow of liquid along the wall.

(vi) Fog, Dispersed, spray or mist flow: Most of the liquid is entrained as a spray in the gas.

Figure 15 shows a sequence of flow patterns which occur in a vertical heated tube, as more and more liquid is converted to vapor (13).

Fog flow is a specific type of two-phase flow having characteristics quite different from those of Bubbly flow, Plug flow, Slug flow, and Annular flow. Very recently Fog flow has become centre of attraction for investigations because it has been recognised as a suitable coolant in Nuclear Reactors e.g. water moderated (light and heavy water)& graphite moderated reactors (3).

Fog is formed of water droplets (whose dimensions are of the order of microns) dispersed in steam, rather than steam in water as in the case of the flow of boiling water.

Fog flow occurs under certain conditions of mass velocity and quality as depicted in Baker's Plot as shown in Figures 1 and 2 (1).* In a diabatic flow, qualities above those of nucleate boiling, the steam voids become large enough to cause discontinuous two-phase flow of steam and water slugs in a narrow channel. At higher qualities the steam phase becomes continuous and at high mass velocity causes a climbing annular film of water on the channel wall and a partial dispersion of water as droplets in the steam. At high enough mass velocity (greater than 5×10^{4} lb/hr-ft^{2}) waves are caused in the film from which additional droplets break off and get

* Number or numbers in the bracket denotes the reference number cited at the end.

dispersed in the vapor stream. This is a typical fog flow.

Fog can also be formed by injecting water droplets into steam. This arrangement is used in nuclear reactors.

The size of droplets dispersed in the vapor depends on the temperature and mass velocity of the two-phase flow. For steam water fog flow at 1,000 lb/sq.in. pressure and 1×10^6 lb/hr-sq.ft. mass flow rates, average droplet size is estimated to be of the order of 20 to 40 microns with a maximum droplet diameter of about 100 to 200 microns (2).

Under adiabatic conditions (i.e. when energy transfer as heat is not taking place across the boundary of the fog flow system) the channel walls are believed to be covered with a thin flowing liquid film. The water film is about 10 mils (25 microns) at room temperature and about 2.5 microns at higher temperatures(3). As heat is transmitted to the coolant through the channel walls the liquid film is believed to break up and disappear at the critical value of heat flux - known as Burnout Condition. Just before the burnout point heat transfer coefficient of the order of 20,000 to 40,000 Btu/hr-sq. ft.°F are obtained - which is characteristic of fog flow.

1.1C <u>Advantages of fog as reactor coolant</u>

Some of the advantages of fog coolant are same as boiling water coolant, and others are unique to fog coolant. Some of the important advantages of this coolant are listed below with reference to nuclear reactors.

(i) Coolant density is much lower than that of liquid or boiling water which reduces the parasitic absorption of neutrons considerably.

(ii) Thermal stresses are reduced because heat transfer to the coolant is by a constant temperature process (evaporation of the liquid) (3).

(iii) Direct admission of coolant into the turbine (after separation of water) simplifies the plant and improves its operating efficiency and eliminates the intermediate heat exchangers.

(iv) Very high heat-transfer coefficients (20,000 to 40,000 Btu/hr-sq.ft.-°F) below burn out are provided at moderate temperatures of the can. (2)

(v) In light water moderated reactors fog-cooling is superior to heat removal by pressurized water and boiling water. For example to produce the same turbine steam conditions, the fog cooled reactor needs much less pressure than the pressurized water reactor. To produce steam in a fog - cooled reactor that has the same properties (560 Psi and 400 °F) as that in Shippingport power station, the primary-circuit pressure would be only 770 Psi, compared with 2,000 psi' in Shippingport.

(vi) Its application is being considered in fast reactors, wher minimal moderation is required in combination with hi· transfer rate. Although fog may not be as good a as medium from these standpoints, its relatively a

forward technology and direct cycle applicability makes it of considerable interest. The film heat transfer coefficient of fog flow is comparable to liquid sodium. For liquid sodium at 700 °F flowing at the rate of 17 ft./sec. through a tube of 1/2 inch inside diameter, the film heat transfer coefficient is around 20,000 Btu/hr. ft.^{2} °F whereas for fog flow under similar conditions it can be around 15,000 to 20,000 Btu/hr. ft.^{2} °F depending upon the quality (2).

(vii) The most important application of fog flow is in Nuclear Super Heat Project. The steam from P.W.R (Pressurised Water Reactor) and B.W.R. (Boiling Water Reactor) is more or less like a fog (wet steam); so a fog cooled reactor can be designed in conjunction with P.W.R. or B.W.R such that the quality of steam can be improved (superheated). This is called the Nuclear Superheat Project and work is being done in U.S.A.

The problems faced in the development of the fog-cooling of reactors are as follows (3) :

(i) Fog must be produced at the inlet of each reactor channel to ensure uniform dispersion of the coolant.

(ii) Coolant velocity over 100 ft./sec. (3) poses a serious erosion problem, which is a characteristic velocity for fog flow.

(iii) Stability of the coolant flow depends on its quality and mass flow rate. At low quality and/or low flow rate, the

coolant flow pattern changes from fog to annular with the bulk of the liquid flowing near the wall.

(iv) Lack of the basic understanding of the phenomena of two phase flow (fog flow).

1.2 Purpose and Scope of the Present Work:

The heat transfer and the fluid flow phenomena (hydrodynamics) in two-phase flow are studied intensively the world over. Much of the previous work has been conducted on single component two phase flow systems and very little data is available on fog flow. The available data mostly pertains to systems simulating high pressure fog cooled reactors.

The purpose of the present experimental work is summarized below:

(i) To procure data on heat transfer, pressure-drop and void characteristics of fog flow in vertical annular channel (simulating the reactor coolant channel geometry) at low pressures. However as the experiment proceeded, some unusual complications arose in heating and measurement of temperature on the inner tube of the annulus and therefore, though some data on temperature distributions have been procured, they were not sufficient to make any statement about the heat transfer characteristics of the flow. And, thus, finally attention was concentrated on pressure-drop and void characteristics of the flow.

(ii) The other purpose of the work is to check whether some of the recent correlations (2,4,8) proposed for fog flow in round tubes at high pressures can be meaningfully extended to low pressure fog flow in annuli.

(iii) From the available literature the author finds that the bulk of the reported works are mainly for two-phase flow studies (Annular flows, Slug flows etc.). The purpose of the present work is also to check whether the correlations and models proposed for two phase studies (4,11,7,8) can be extended to predict the behaviour (pressure drop, void-fraction distribution and heat transfer coefficient) of fog flow (which is a particular case of two-phase flow).

The range of experimental variables covered here is as follows:

Total mass flow rates	: 2.21×10^5, 2.3×10^5, 2.66×10^5 lbs/hr. ft^2
Qualities	: 0.40 to 0.60
Inlet Pressures	: 85 Psia and 100 Psia
Annular Pipe Sizes	: Inner Tube diameter = 1/2" (O.D)
	Outer Tube diameter = 1" (O.D)
	Wall thickness = 1/16"

CHAPTER II

THEORETICAL CONSIDERATIONS

2.1 General :

From an engineering view point the final objective of a study of two-phase flow (fog or any other) is to determine the pressure drop, void distribution and heat transfer characteristics. The theory concerning the flow of two phases is not well known because of some inherent complications of such flows. The reasons of the complications can be attributed to the following facts (5, 6).

(i) Two-phase flow with heat addition is a coupled thermohydrodynamic problem. On the one hand heat transfer causes phase change and hence a change in the distribution of phases and flow patterns; on the other hand, the change in hydrodynamics as caused by pressure drop along the flow path affects the heat transfer characteristics.

(ii) Single component two-phase flow in a vertical or horizontal conduit can never become fully developed because the inherent pressure changes along the conduit continually changes the state of the fluid and thereby changes the phase distribution and flow patterns.

(iii) Possible occurrence of metastability in the flow i.e. possibility of the lack of thermodynamic equilibrium.

(iv) Presence of 'Bernoulli Effect' in the flow (5). The Bernoulli's Effect is a speeding up of the less dense phase relative to the more dense phase in an accelerating flow. Exposed to the same pressure gradient the less dense phase attains higher kinetic energy and hence a higher velocity than the more dense phase. The difference in the two velocities is called 'Slip'. Slip reduces the momentum and kinetic energy of both phases associated with a given mass velocity (6).

(v) Possibility of a large number of flow patterns.

2.2 Theories for Pressure Drop Predictions :

Despite the complications mentioned above many simplified analytical model studies have been conducted with some success.

(a) Homogeneous Model (7):

Here the components are treated as a pseudo fluid with average properties without bothering with a detailed description of flow patterns. This pseudofluid obeys the usual equations of single component flow. This model is applicable for a fog or spray flow patterns occuring at high void fractions.

The basic assumptions of a homogeneous model are (5) :

(i) Equal linear velocity of vapor and liquid.

(ii) Thermodynamic equilibrium between the two phases.

(iii) A suitably defined single-phase friction factor is applicable to the two-phase flow.

The basic equations for steady one-dimensional homogeneous equilibrium flow in a duct are :

Continuity : $W = \rho_m v A = \text{Constant}$ (2-1)

Momentum : $W \frac{dv}{dz} = - A \frac{dp}{dz} - P \tau_w - A \rho_m g \cos\theta$ (2-2)

Energy : $\frac{dq}{dz} - \frac{dw}{dz} = W \frac{d}{dz} \left(h + \frac{v^2}{2} + g z \right)$ (2-3)

Here $\frac{dq}{dz}$ represents heat transfer per unit length of the duct, z is the vertical co-ordinate, θ is the inclination of the duct to the vertical.

Equation (2-2) is often written in an explicit form for pressure gradient. Thus

$$\frac{dp}{dz} = - \frac{P}{A} \tau_w - \frac{W}{A} \frac{dv}{dz} - \rho_m g \cos\theta \qquad (2\text{-}4)$$

The three terms on the right side can be regarded as frictional, accelerational and gravitational components of the pressure gradient. Engineers, mostly interested in pressure drop study, usually adopt the following definitions :

$$- \left(\frac{dp}{dz} \right)_F = \frac{P}{A} \tau_w \qquad (2\text{-}5)$$

$$- \left(\frac{dp}{dz} \right)_A = \frac{W}{A} \frac{dv}{dz} \qquad (2\text{-}6)$$

$$- \left(\frac{dp}{dz} \right)_G = \rho_m g \cos\theta \qquad (2\text{-}7)$$

The total pressure gradient is the sum of three components, as follows :

$$\left(\frac{dp}{dz}\right)_{total} = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_A + \left(\frac{dp}{dz}\right)_G \qquad (2\text{-}8)$$

The mean properties of flow can be defined in several ways and the equations can further be developed in terms of easily measurable quantities. For example the mean density can be expressed as follows:

$$\rho_m = \alpha . \rho_g + (1 - \alpha)\ \rho_l \qquad (2\text{-}9)$$

or

$$1/\rho_m = \frac{x}{\rho_g} + \frac{1 - x}{\rho_l} \qquad (2\text{-}10)$$

Where α and x are void fraction and quality of the flow at a particular cross-section, and they are defined as

$$\alpha = \frac{A_g}{A_l + A_g} \qquad (2\text{-}11)$$

$$x = \frac{W_g}{W_l + W_g} \qquad (2\text{-}12)$$

It is to be noted that this quality is not a local mean value in stationary sense but a local value weighted by local mass flow rates.

The static quality x_s is defined as the ratio of the weight of vapor in a section of pipe to the total weight of the fluid in that section.

Thus the relation between x and x_s is given by (9) -

$$\frac{x}{1 - x} = S\ \frac{x_s}{1 - x_s} \qquad (2\text{-}13)$$

Where S is the Slip Ratio defined as

$$S = \frac{V_g}{V_f} = \left(\frac{V_2}{V_1}\right) \tag{2-14}$$

(i) One of the best schemes based on homogeneous model has been proposed by Owens (7), who defines the mean specific volume as follows:

$$\bar{v} = \frac{W_2 v_2 + W_1 v_1}{W_2 + W_1} = v_1 \left(1 + \frac{x}{v_1} (v_2 - v_1) \right) \tag{2-15}$$

Now the momentum equation can be developed further by expressing the wallshear forces in terms of a friction factor and a hydraulic mean diameter D_e. The average wall shear stress is

$$\tau_w = f_{TP} \cdot \frac{1}{2g_c} \cdot \frac{V^2}{\bar{v}} \tag{2-16}$$

Where f_{TP} is two phase friction factor

$$- \left(\frac{dp}{dz} \right)_F = \frac{f_{TP} \, G^2 \, \bar{v}}{2 \, D_e \, g_c} \tag{2-17}$$

Also

$$- \left(\frac{dp}{dz} \right)_A = \frac{G^2}{g_c} \frac{d\,\bar{v}}{dz} \tag{2-18}$$

and

$$\left(\frac{dp}{dz} \right)_G = \frac{1}{\bar{v}} \; \frac{g}{g_c} \tag{2-19}$$

Where $\bar{v}$ is specific volume defined above

Hence

$$\left(\frac{dp}{dz} \right)_{Total} = \frac{\dfrac{f_{TP} G^2 v_1}{D_e \, 2g_c} \left[1 + x \left(\dfrac{v_2}{v_1} - 1 \right) \right] + \dfrac{G^2 v_1}{g_c} \left(\dfrac{v_2}{v_1} - 1 \right) \dfrac{dx}{dz} + \dfrac{g}{g_c v_1 \left[1 + x \left(v_2/v_1 - 1 \right) \right]}}{1 + x \dfrac{G^2}{g_c} \quad \dfrac{d\,v_2}{dp}} \tag{2-20}$$

Owens has suggested that the friction factor of the single phase fluid flow should be used for f_{TP}.

(ii) Sofer's Modified Homogeneous Model (2) :

Based on the data of frictional pressure drop Sofer has reported a correlation using homogeneous flow model approach employing an empirically modified Reynold's number. This is specially applicable to fog flow. In this case

$$\Delta P = 2 f_{TP} \frac{L}{D} \bar{v} G^2 + (\bar{v}_e - \bar{v}_i) G^2 \qquad (2\text{-}21)$$

Where Δp is pressure drop in a length L of the channel

$$f_{TP} = 0.046 \quad \frac{DG}{\bar{\mu}} \left[\left(\frac{G}{1.77 \times 10^6} \right)^2 \right]^{-0.2} \qquad (2\text{-}22)$$

$$\bar{v}_e - \bar{v}_i = v_{12} (x_e - x_i) \qquad (2\text{-}23)$$

$$\bar{v} = x v_2 + (1 - x) v_1 \qquad (2\text{-}24)$$

$$\bar{\mu} = x \mu_2 + (1 - x) \mu_1 \qquad (2\text{-}25)$$

Here subscripts e and i refers to outlet and inlet properties.

He has tested this correlation in high pressure fog flow (steam - water at 1,000 psi) and found good agreement in the case of adiabatic flows in round tubes.

(b) <u>Slip Model or Separated Flow Model</u>:

This model takes account of the fact that the two phases can have differing properties and different velocities. It may be developed with various degrees of complexity. In the most sophisticated version, separate equations of continuity, momentum and energy are written for each phase and these six equations are solved simultaneously, together with the rate equations which describe how the phases interact with each other and with the walls of the duct. In the simplest version, only one parameter, such as velocity is assumed to be different for the two phases while conservation equations are written for combined flow.

This model is applicable for annular flows and fog flows having intermediate values of void fraction.

The equations of motion can be expressed as :

Continuity : $W = W_1 + W_2 = \text{Constant}$ (2-26)

Where

$$W_1 = \rho_1 V_1 A_1 \tag{2-27}$$

$$W_2 = \rho_2 V_2 A_2 \tag{2-28}$$

Momentum :

Many alternative forms of the momentum equation can be derived by manipulating relationships among α, x, G, V_1, V_2 and other variables. For steady flow in a round pipe, for example, one version is (8) :

$$-\frac{dp}{dz} = \frac{4\tau_w}{D} + G\frac{d}{dz}\left[xv_2 + (1-x)\,v_1\right] + \left[\alpha\rho_2 + (1-\alpha)\,\rho_1\right]\cos\theta \tag{2-29}$$

Energy :

The energy equation is conveniently written in terms of the quality. Thus :

$$\frac{1}{W}\left(\frac{dq}{dz} - \frac{dw}{dz}\right) = \frac{d}{dz}\left[xh_2 + (1-x)h_1\right] + \frac{d}{dz}\left[x\frac{v_2^2}{2} + (1-x)\frac{v_1^2}{2}\right] + g\cos\theta \tag{2-30}$$

To solve the above set of equations, two further relationships are required in addition to the relationships between the thermodynamic properties. The best way to derive these two relationships is by analysing the motion of each component (or phase) separately; but this becomes a complicated task. The common technique is to use empirical relations for τ_w and α in terms of flow rates, fluid properties and geometry. One of the widely used correlation, in this context, is that of Martinelli et. al. (10) and Martinelli - Nelson (11).

Lockhart and Martinelli (10) recognised four types of flows based on whether the liquid phase flow is turbulent or gas phase flow is turbulent -

(i)	Turbulent	-	Turbulent	(t - t)
(ii)	Viscous	-	Turbulent	(v - t)
(iii)	Turbulent	-	Viscous	(t - v)
(iv)	Viscous	-	Viscous	(v - v)

They gave correlations for each type of the flow. However the flow with practical importance is the first one only (t - t).

The expression for the two-phase friction pressure drop in terms of single phase pressure drop of gas flow alone is (10)

$$\left(\frac{dp}{dz} \right)_{TPF} = \left(\frac{dp}{dz} \right)_{GPF} \phi^2_{gtt} \qquad (2\text{-}21)$$

Where ϕ^2_{gtt} is an empirical function of X_{tt} which is a dimensionless correlation parameter obtained by dimensional analysis (10) and defined as

$$X_{tt} = \left(\frac{(dp/dz)_{LPF}}{(dp/dz)_{GPF}} \right)^{\frac{1}{(2-n)}} \qquad (2\text{-}22)$$

Rearranging (2-22) gives

$$X_{tt} = \left(\frac{\rho_2}{\rho_1} \right)_{sat}^{\frac{1}{(2-n)}} \left(\frac{\mu_1}{\mu_2} \right)^{\frac{n}{(2-n)}} \left(\frac{1-x}{x} \right) \qquad (2\text{-}23)$$

Where n is determined empirically from $f = C/Re^n$ and $n = 0.20 - 0.25$ for a turbulent flow. When a two-component flow is considered, the subscript "sat" for the density ratio refers to the saturation temperature and pressure of the liquid flow.

These results were later modified by Martinelli and Nelson (11) to the following expression for steam - water flow :

$$\frac{\left(\frac{dp}{dz} \right)_{TPF}}{\left(\frac{dp}{dz} \right)_{LO}} = \phi^2_{ltt} \left(1 - x \right)^{1.75} \qquad (2\text{-}24)$$

Then the local value of $(dp/dz)_{TPF}$ / $(dp/dz)_{LO}$ was plotted against local quality x and pressures as shown in the Figures 5 and 6.

2.3 Theories for Void - Fraction Prediction :

The accurate prediction of void fraction (volumetric steam fraction α) in water - steam mixture is essential to the design of fog - cooled or water moderated reactors and to a complete understanding of the hydrodynamic characteristics of two-phase flow (12). However, the status of theoretical analysis for this problem is same as that of the pressure drop. Quite a few theoretical models have been proposed to render the problem amenable to mathematical treatment. Before describing the different models, it is necessary to study the basic concepts related to void fraction analysis :

Void fraction α is defined as

$$\alpha = \frac{A_2}{A_1 + A_2} \tag{2-11}$$

The Slip Ratio S is defined as

$$S = \frac{V_2}{V_1} \tag{2-14}$$

The flow quality is defined as

$$x = \frac{W_2}{W_1 + W_2} = \frac{A_2 G_2}{A_2 G_2 + A_1 G_1} \tag{2-12}$$

From these definitions it follows that

$$S = \frac{x}{1 - x} \cdot \frac{1 - \alpha}{\alpha} \cdot \frac{\rho_1}{\rho_2} \tag{2-25}$$

As seen from the equation (2-25) the void fraction may be evaluated when the slip-ratio of the flow is known, assuming that phases are in equilibrium and that quality is known (either by measurement or by heat balance).

The slip ratio is usually obtained from experimental data through correlations, though there are a few analytical expressions based on theoretical models of the flow.

(i) <u>Homogeneous Model</u> :

This is the simplest of all models and theories for void fraction prediction, whose basic features have already been described.

This is applicable for fog-flow at high void fraction and assumes that slip is equal to unity. Thus, from equation (2-25)

$$\frac{\alpha}{1-\alpha} = \frac{x}{1-x} \cdot \frac{\rho_1}{\rho_2} \qquad (2\text{-}26)$$

The graphs for α vs x have been plotted for a steam water fog at 100 psia and 25 psia in Figures 3 and 4.

(ii) <u>Levy's Momentum Exchange Model</u> (4)

It is postulated here that momentum is exchanged between liquid and vapor whenever quality, void fraction or density ratio varies and that this exchange tends to maintain the sum of the frictional and head losses equal for the two phases.

Writing the fundamental momentum equations for each phase (of Bernoulli type) and then applying the above postulates, Levy derived that (for horizontal or vertical flow) -

$$\frac{(1-x)^2}{(1-\alpha)} + \frac{x^2}{\alpha} \cdot \frac{\rho_1}{\rho_2} - \frac{1}{2} \frac{(1-x)^2}{(1-\alpha)^2} = 0 \qquad (2\text{-}27)$$

From here x can be solved in terms of α .

At low pressures where the quantity (1-x) may be set equal to unity even at high void contents, the equation reduces to

$$x^2 = 2 \frac{\alpha^2}{(1-\alpha)^2} \frac{\rho_2}{\rho_1} \qquad (2\text{-}38)$$

and the slip ratio is

$$S = \frac{V_2}{V_1} = \left(\frac{\rho_1}{\rho_2}\right)^{1/2} (2\alpha)^{1/2} \qquad (2\text{-}39)$$

Thus the slip ratio varies as $(\rho_1/\rho_2)^{1/2}$, where as in homogeneous model the dependence is on ρ_1/ρ_2 . The author reports that the model gives good agreement with available experimental results in horizontal and vertical test sections with and without heat additi0n at pressures from 12 to 2000 psia.

The plot of α vs x derived from this model is shown in Figures 3 and 4.

CHAPTER III

LITERATURE SURVEY

3.1 General:

The current status of knowledge and practice in two - phase flow, in general, has been well summarised in the following paragraph by Dukler in A.I.Ch.E. Journal (Jan. 64) (19) :

"Multiphase flow still suffers when compared on a theoretical basis with other general fields of flow theory as is natural in view of the great complexity of the problem. We may note, however that, the build up of empirical knowledge in this field is now very impressive. The total number of experimental measurements of two - phase pressure drop is currently well above 20,000 half of which have been obtained since 1960. It is evident that two - phase flow is a timely subject. Furthermore this continued accumulation of data demonstrates that there is not yet even a phenomenological under - standing of this type of flow."

The literature in this field is quite extensive as can be seen from a recent annotated bibliography which cites over 25,000 references for the period from 1950 to 1962 (20). Further a general index of 2368 references on two-phase flow has been compiled by Gouse (21).

However the status of literature in steam - water fog flow is not very encouraging. Significant number of installations are engaged in the study of this area. Three companies are jointly studying fog flow with special reference to its applicability to water moderated reactors (3).

(a) Centro Informazioni Studi Esperienze (C.I.S.E.) in Milan, Italy where team headed by M. Silvestri is carrying out research programmes of fog flow with respect to heat transfer, flow stability and corrosion - erosion.

(b) Ansaldo Company, Genoa is developing special injection nozzles for producing fog at proper quality at the base of the reactor coolant channel. Work is in progress on a large high-temperature high pressure loop for this purpose.

(c) In Nuclear Development Corporation of America (NDA) work is in progress on the detailed analysis of the applicability of fog cooling to light - water - moderated reactors.

(d) In the Atomic Energy Research Establishment (A.E.R.E.) of U.K. work is being done on the fundamental character of the behaviour of steam - water fog flow (13).

(e) Atomic Energy of Canada Ltd. (AECL) is engaged in the study of fog - coolant in heavy water moderated, natural uranium reactor (3).

In this and the subsequent chapters some of the references will be reviewed.

3.2 Flow Pattern Studies :

The initial problem of interest is that of establishing those fundamental variables which determine the nature of flow in two-phase systems (20). There is some variation in the definition of the flow patterns given by different observers. Attempts have been made to bring some order to the terminology. Bradley (20) has reported the comparison of the works of eight authors in terms of types of flow and has diagrammatically depicted the interrelationship of flow pattern terminologies. Figure 15 reproduces a diagram (13) which has been found convenient for depicting most of the possible flow regimes in upward co-current (both phases moving in the same direction) vertical two-phase flow. The wall is subjected to constant heat flux. The change from nucleate boiling to fog flow (section E plus F) via 'slug' and 'churn' flow is essentially a phase change; the upper boundry of the "liquid - dispersed" region (F) is where the dry out conditions start.

Within the fog region large changes are discernible. At low flow rates all the water may flow in the film (annular flow E); at high flow rates nearly all of it flows in form of suspended particles (Fog, F). Waves are present in all cases at the liquid film - vapor interface (13).

Despite the lack of data, it is possible to construct a flow pattern chart for steam and water at low and high pressures based on visual observation at low pressures of air -water flow.

One such chart has been prepared by Sedmann et al (22) using Baker's (23) flow pattern chart as modified by Isbin et al (24). The various flow - patterns are indicated as functions of the quality (x %) and the total mass velocity (G lbs/hr.ft.^{2}) for a pressure of 25 psia, 100 psia, as in Figure 1 and Figure 2.

2.3 <u>Hydrodynamic Studies</u>:

Because of complications in the mathematical models required for theoretical analysis of the problem the more successful approaches have been empirical and semi - empirical. Two principal types of flow models have appeared in literature in the analysis of pressure drop and void fraction - Homogeneous Model (7, 20) and the Slip Model (10, 11, 4, 21).

The details of Owen's work (7) were given in Chapter 2. The agreement of this model with experiments have been reported to be good.

Mendler (20) used a mean specific volume defined by weighting with flow rates and he used a liquid single phase friction factor for calculating pressure loss at 600 - 1800 psia. Here natural and forced convection test data for a closed - loop system were presented and analyzed. The data were obtained at pressures of 600, 1200, 1600 and 2000 psia from the natural circulation loop, using rectangular channel test - sections (.100 in. x 1.0 in. x 27 in. long, .200 in. x 1.00 in. x 27.0 in. long and .300 in. x 1.0 in. x 27.0 in long).

The results showed that single and two phase pressure drop, burn out heat flux and riser density measured under natural circulation was no different from those measured with forced circulation for the same fluid flow conditions. Some data for slip ratios were obtained. Agreement of his correlation with the other existing correlations was found good.

Levy (4) has postulated "a momentum exchange model" such that the exchange of momentum between liquid and vapor tends to maintain the sum of frictional and head losses equal for the two phases. Giving mathematical expression to this statement he derived Bernoulli type momentum equations from which he gets the expression for two phase pressure drop

$$(dp/dz)_{TPF} = (dp/dz)_{LO} \quad \frac{(1-x)^{1.75}}{(1-\alpha)^{2}}$$

It is to be noted that the expression is independent of mass velocity G and the geometry of the channel D_e. This model gives good agreement with experimental results (4) in horizontal and vertical test sections (round test sections) with and without heat addition at pressures from 12 to [illegible] psia. Predictions are [illegible]% below the values measured. Bankoff ([illegible]) reports that the slip velocities predicted by this model are higher than experimental values and void fractions predicted are lower than experimental values.

Marchaterre (31) has suggested a modification to Levy's model that incorporates the mass velocity "G" and diameter effect into the pressure drop equation. Assuming that the liquid phase

pressure drop can be expressed in terms of some apparent liquid friction factor, f_1, and equivalent pipe diameter $D_{e(1)}$, he derives

$$\left(\frac{dp}{dz} \right)_{LPF} = - \frac{2 f_1}{D_{e(1)}} \cdot \frac{G^2}{\rho_1 g_c} \frac{(1-x)^2}{(1-\alpha)^2}$$

Using Levy's postulate and the relation

$$\left(\frac{dp}{dz} \right)_{TPF} = \left(\frac{dp}{dz} \right)_{GPF} + (1-\alpha) \left(\frac{dp}{dz} \right)_{LPF}$$

he obtained the expression for $\left(\frac{dp}{dz} \right)_{TPF}$ which includes G and D_e. These (Levy's and Marchaterre's) models were compared with the experimental observations of pressure drop on round tubes and annular tube for vertical flow :

Round Tube : (a) I.D. = 0.80 cm., Pressure 70 kg/cm^2 and G = 114 to 487 gm/cm^2 sec.

(b) I.D. = 1.01 cm., Pressure = 70 kg/cm^2 and G = 110 to [illegible] gm/cm^2 sec.

Annular Tube: I.D. = [illegible] cm., O.D. = [illegible] cm., De = [illegible] cm., Pressure = 70 kg/cm^2 and G = [illegible] to [illegible] g/cm^2 sec.

Comparison of the experimental data with Levy's correlation shows limited agreement for the range of variables considered. This is attributed to the fact that this correlation does not take into account a dependence on mass flow rates and diverges as the quality approaches one.

The agreement between Marchaterre's correlation and the data is poor due mainly to the large dependence of this correlation

on the tube diameter. This dependence is not so noticeable in the experimental results.

Lockhart and Martinelli (10) presented a moderately successful correlation for two - component flows and this was further extended by Martinelli and Nelson (11) for the case of single component (steam - water) flows. Their correlation was based on the data obtained for annular flow patterns. In this correlation the pressure drop during two phase flow is related to the pressure drop occuring if only a single phase flowed in the pipe. The correlations are in the form of graphs of ϕ^2_{LO} versus quality, the pressures being parameters. Some curves are shown in Figures 5 and 6A at pressures 85 psia and 100 psia respectively.

Separated flow model was used for correlating the data (this correlation and model has been described in Chapter II). Using this model they expressed the two - phase pressure drop in terms of the pressure drop if only one phase were flowing in the tube.

$$\left(\frac{dp}{dz} \right)_{TPF} = \left(\frac{dp}{dz} \right)_{LPF} \quad \phi^2_{ltt}$$

It was empirically established that

$$\left(\frac{dp}{dz} \right)_{LPF} = \left(\frac{dp}{dz} \right)_{LO} \left(1 - x \right)^{1.75}$$

where

x : local quality

LPF : refers to superficial liquid flow rate i.e. when only liquid component rate is flowing as liquid,

LO : refers to actual liquid flow rate i.e. when the total flow rate of two phase flow is flowing as liquid.

From their curves of ϕ^2_{LO} versus quality at different pressures, the pressure drop in two phase flow is given simply by multiplying ϕ^2_{LO} with the liquid phase pressure drop, if the flow is adiabatic. For the diabatic case, a further assumption is made about the linearity of quality along the channel length, and then ϕ^2_{LO} is plotted against the exit quality. After knowing the quality and pressure of the flow, ϕ^2_{LO} is known from their curves and by multiplying this with the single phase liquid pressure drop, the pressure drop in the two - phase flow can be calculated. This correlation is a widely accepted one and the experimental results of the present investigation will be compared with the predictions of this correlation.

A correlation developed by Chenoweth and Martin (24) has been successful in correlating data for large pipe sizes (3 in. I.D. and over). The investigation was undertakento check the Lockart and Martinelli (10) correlation (of pressure drop) at higher pressures (over 100 psia) and larger pipe sizes (over 3 in. I.D.). The largest deviations (by a factor of 1.4 to 2.5) were observed at 100 psia in 3 inch I.D. pipe.

Buckler et. al. (19) have systematically compared the existing correlations of two - phase flow frictional pressure drop data and void - fraction variations. A data bank was prepared of about

9000 data points covering wide range of experimental parameters like flow patterns, mass - velocities, pressures, geometry of the channel, heat - flux, quality of the flow etc. After careful culling, about 2520 data points were left for testing various correlations. Out of the 25 correlations that appeared in the literature at that time, the five (which found wide use) were chosen : Baker (1), Bankoff (36), Chenoweth and Martin (34), Lockhart and Martinelli (10) and Yagi (37) for testing with the data. Overall comparison showed that Martinelli Nelson correlation fits best to the data. Comparison of Void - fraction correlations with data showed poor performance for all the correlations tested.

The correlation proposed by Marchaterre and Hoglund (35) for void - fraction in vertical two - phase flow agrees with the data within 20%. They have presented a correlation for predicting vapor volume fraction in vertical channel of Boiling Water Reactor. They have picked Froude Number, the velocity ratio and the ratio of the volumetric flow rate as significant dimensionless groups and have presented their correlation in the form of a graph.

Bankoff (36) proposed a variable density single fluid model for the analysis of two - phase (steam - water) flow. His assumptions are :

(i) Mixture flows as suspension of bubbles (void) in the liquid and radial gradient exits in the concentration of bubbles.

(ii) The bubble (void) concentration is maximum at the centre of the pipe, decreases monotonically in a radial direction and

vanishes at the pipe wall i.e. pipe wall is wetted with the liquid only.

(iii) Vapor and liquid have same velocity at any radial position, i.e. the slip is considered to be negligible compared to the steam velocity.

(iv) Thus the model is essentially a single fluid model whose density is a function of radial position.

(v) Power law distribution is asserted for velocity and void:

$$u^* = u/u_m = S^{1/m}$$

$$\alpha^* = \alpha/\alpha_m = S^{1/n}$$

and he derived, $\bar{\alpha}/\alpha_m = 2n^2 / (n+1)(2n+1)$

where $S = y/R$, R = radius of the tube,

$\bar{\alpha}$ = Average void fraction on the cross section.

Subscript m refers to the conditions at the tube centre.

He defined a parameter K as

$$K = \frac{2\,(m + n + mn)\,(m + n + 2\,mn)}{(n+1)\,(2n+1)\,(m+1)\,(2m+1)}$$ which varies from 0.6 to 1.0. For an assumed value of K = 0.89, good agreement was reported with Martinelli - Nelson (11) correlation for steam - water void fraction and frictional pressure drop over a range of pressures from 100 to 2000 psig and void - fractions from 0 to 0.80. Dukler (10) has reported that this correlation is better for single component flows than for two components and is better at higher pressures.

Sedor (8) has presented a good account of the merits and limitations of fog flow as reactor coolant. At the same time he has given the highlights of the accepted methods of analysis and prediction of flow pattern stability, pressure drop, void fraction variation, heat transfer and critical heat flux characteristics. He has used homogeneous flow model and has proposed an empirically modified Reynold's number (thus a modified friction factor) to predict the pressure drop. This correlation was found good for adiabatic flow in round tubes, but the experimental data in annuli, and in round tube under heat addition were predicted within a factor of 2. All these tests were conducted at and above 800 Psia pressures.

An interesting observation which was made here on heat transfer under fog flow conditions is the fact that a step increase in the heat transfer coefficient occurs reproducibly as the heat flux is increased to about 70% of the critical value (burn out value). Decreasing the heat flux at this point does not cause a step decrease in the heat transfer coefficient but causes higher coefficients than those obtained prior to the step rise. Thus the heat transfer coefficient below burn out appears to be subject to a so called "hysteresis effect" and reverts to the lower value only after the heat flux has been decreased below 30% of the critical value. This observation may shed light on the mechanism of the heat transfer in fog flow.

Shiv Kumar (14) has experimentally investigated the variation of void - fraction and slip ratio with pressure and weight fraction in vertical two phase air - water flow. He has presented

in detail, the experimental procedure of Gamma - ray attenuation method for measuring void - fraction along with its theory. He used an aluminium pipe of square cross - section (3.22 cm square internal) as test channel. Parameter ranges studied are pressure 3.0 psig to 12.5 psig and air - weight fraction from 0 to 5.6%. He concluded that void fraction increases with pressure and air - weight fraction due to decrease in slip ratio at higher pressure. The distribution of void in a cross - section shows a definite tendency for the voids to collect near the centre.

His observation of increase of void fraction with pressure is in contradiction with our observations in the single component fog flow.

Zuber and Findlay (41) derived a general expression which can be used for predicting the average volumetric concentration of steam in two phase steam water flow. The analysis takes into account the effect of the non - uniform flow and concentration distribution across the duct as well as the effect of the local relative velocity between the two phases. The results are general and can be applied to any two phase flow regime. The weighted mean gas velocity is related to the total flow rate of the mixture by means of a constant which lies between 1.2 and 1.5

A detailed survey of literature in this field has been given by Collier (28) upto 1967. The most recent survey of the literature (upto 1969) has been compiled by Cousin (30) and Kutateladze (40).

3.4 Criticism of the Earlier Investigations :

1. From the presented literature survey one can easily see that the 'models' proposed by the different investigators for the analysis (and developing correlations for pressure drop void - fraction, flow patterns etc.) of two - phase flow suffers from atleast one defect - "All of them are either one extreme case or the other and are thus unrealistic models".

I shall justify my statement by first recognising the fact that all the 'models' and 'correlations' proposed for two - phase flow studies can be conveniently grouped in one of the three basic groups :

i) Homogeneous Model Group, where the two components (and phases) in the flow are assumed to be homogeneously mixed such that they are considered 'One Fluid - a pseudo - fluid - having some average properties'. It is not hard to see that this is far from the actual phenomena and thus this model is one extreme (limiting) case of the real situation. The reported work of Owens (7), Mandler et. al. (30) Sefar (2), Wallis (26), Harvey and Foust (32), Linning (33) etc. are different variations of this model.

ii) Separated Flow Model (Slip Model) Group: This is the other extreme (limiting) case of the actual phenomena where it is assumed that the two components and phases are such that one slides over the other at their common surface.

It assumes that each of the two phases are completely separated which is certainly too much to expect. The reported works of Lockhart and Martinelli (10), Martinelli Nelson (11), Levy (4), Marchaterre (31), Marchaterre and Hoglund (35), Isbin et. al (34), Yagi (37) etc. are the different variations of this model.

iii) The Variable Density Single Fluid Model Group : This model makes an attempt to compromise the above mentioned two extreme cases by assuming that the two - phase mixture can be taken as one fluid (no slip between the two phases) with its associated average properties and at the same time it is also assumed that the density of this fluid varies radially at a particular cross - section, starting with a pure gas at the centre of the channel to the pure liquid at the wall of the channel. Bankoff (36) is the propounder of this model and though this model has more mathematical complications, it appears to have more promise, being closer to the actual phenomena than the other two. Zuber et. al (41) have extended this work further.

2. It is easy to see that inspite of the large amount of literature in the area of the study of two - phase flow, the information is still scarce on fog - flow which is a special case of two-phase flows. Though the present work is concerned with fog flow, a fairly exhaustive literature survey on two phase flow

has been presented here with a view to sorting out the possibilities of extending the correlations and assumptions of the general two - phase flows to the case of fog - flow.

Also the other point which becomes obvious from the literature survey is that practically no data are available for the analysis of fog - flow at low pressures (100 psia and below).

CHAPTER IV

EXPERIMENTAL

4.1 Experimental Set Up :

The proposed study of two - phase (steam - water) fog flow required the experimental set up to be sufficiently versatile. Hence the set up was designed and fabricated to provide for easy installation of different test sections.

The apparatus, built in the Energy Conversion Laboratory is shown schematically in Figure 8. Views of the equipment are shown in Figures 18 to 21.

As it now stands, the equipment provides means to prepare and feed into the various test sections with a mixture of steam and water or wet steam only (without inter - mixing it with water).

Keeping in view the complications of the equipment each part of the apparatus will be considered separately.

Steam, coming from the boiler, enters the mixing chamber where it gets mixed with water. Water (stored in a tank) is pumped into the mixing chamber (by a multistage pump) in the form of spray (small droplets). The water spray is formed by an atomiser attached

at the end of the water - line in the mixing chamber. Steam - water fog mixture formed in this manner enters at the bottom end of the vertical test channel and then after flowing through the test length it is exhausted to the atmosphere.

Pressure gauge and rotameter measure the pressure and flow rate of water sprayed into the mixing chamber. The pressure gauges, orifice - meter and throttling calorimeter are used for measuring pressures, flow rate and the quality of the steam entering into the mixing chamber.

4.1.1 Steam Supply :

An oil-fired steam boiler was used as the source of steam (saturated and wet). The steam Generator is Kestler Make with a capacity of 3600 lbs/hr at 150 psig.

4.1.2 Water Supply :

Ordinary tap water was stored in a tank 8' x 4' x 4' from which a four stage (high pressure) centripetal pump (capacity 3 gpm at 300 psi pressure) pumps the water to the mixer section through a sprayer. Specifications of the pump and motor are :

Pump : KE - SIHI Model A 06140 K

Motor : Crompton Parkinson, Bombay

1430 RPM 3 Phase 50 Cycles

400/440 Volts 7.5 Amps Each

5 H.P.

4.1.3 Mixer (Figures 12 & 13) :

The purpose of the mixer is to mix the phases (steam and water) so that the discharge entering the test section would be well mixed and be at equilibrium. The unit consists of a conical mixing chamber, a nozzle for spraying the incoming water and an entry port for steam. The nozzle was designed and fabricated in the Central Workshop. The dimensions and arrangement is shown in Figure 9.

The outer body of the mixer is insulated with 1/2 inch thick magnetia coating.

4.1.4 Test Section (Figures 10 & 24) :

An annular test section of circular geometry was designed and fabricated (with the help of Central Workshop and the Precision Shop) such that the pressures, void fractions and temperatures can be convenienptly measured. The details of the test section showing the pressure taps and thermocouple arrangements is shown diagrammatically in Figure 10. The figure shows in detail the angular positions of thermocouple junctions (at a particular cross-section) and positions along the length of the test section.

The test section was built using two stainless tubes of wall thickness of 1/16 inch, outer diameters 1/2 inch and 1 inch and lengths 7 ft. and 6 ft. respectively.

On the 1/2 inch diameter tube twelve splines (1/32" x 1/32") of different lengths were cut on its outer surface to fix the

the thermocouples. Six sections were chosen at 6" apart for temperature measurement and at each section two thermocouple bulbs (diametrically opposed) were peened on the surface. The thermocouple wires were carefully inserted into the splines and then cemented with Sauereisen (imported) Cement.

On the outer tube, again at the same six sections pressure taps - one at each section - and thermocouples - three at each section at 120° angle were fixed. A through hole was made in the tube into which the thermocouple bulb was inserted such that it is just flush with the inner surface of the tube and then it was peened and cemented.

All the thermocouples were tested by heating the inner tube with the heating element before installing the test section to see whether they are working (or got broken) and whether they read the same temperature at a particular cross section.

The annular test section was obtained by 1/2 inch tube put inside the 1" tube andfixing it centrally with the end arrangements (detail is shown in Figure 18).

The pressure taps were designed and fabricated in the Precision Shop and they were silver brazed on the stainless steel tube. They are made of brass. The dimensioned drawing and method of fitting of the pressure taps on the outer tube is shown in Figure 19. The length of the pressure taps is 1-1/8 inch and the outer diameter 3/4 inch. There is 1/8 inch diameter hole through the body

of the pressure tap. 1/2 inch diameter swage locks (imported) were then fitted in these taps for connecting copper piping to the manometer manifold.

Out of the total length of 72" of the test - section, only 36" was test length. Before the fog - mixture entered this test - length, it had to pass through the settling - length of 20". Also the fog - mixture was discharged into a discharge - settling length of 16" after its emergence from the test length.

The whole length of the test section was completely insulated by putting a 3/8 inch thick coating of fire-proof insulation cloth and magnesia, except at the section where the void fraction was being measured. A 1/2" long slit was cut from the insulation on the periphery at the section to give passage to gamma ray beam for void fraction measurement. A full view of the test section with end connections is shown in Figure 24.

4.15 <u>Heating Elements</u>:

The 36" long electrical resistance heating element (imported) of 3/8 inch O.D. and 1 kw capacity was used to heat the inner tube. This element was inserted into the 1/2 inch tube and was fixed in the position such that it could heat the test length of the test section. The power supply was given through a Variac Transformer through ammeter and volt meter.

4.1.6 <u>Piping Valves and Insulation</u>:

Steam line piping was of high pressure m.s. pipe of 1" I.D.

Needle Valves were used on the water and steam lines. Water lines were made up of 1" I.D. and 2" I.D. m.s. pipes. Water lines were not insulated. The steam line, the mixer and the test section were insulated with magnesia pipe lagging and fire - proof cloth giving 1/2 inch thick coating on them.

4.1.7 <u>Cooling Jacket</u> :

The water pump was used to get wetness more than 75%;to get wetness below this value a cooling jacket was installed in the steam line. Its view is shown in Figure 21.

4.2 <u>Instrumentation</u> :

The instrumentation part was most important. It was designed to supply the following measurements :

4.2.1 Flow rate.

4.2.2 Temperature.

4.2.3 Pressure.

4.2.4 Void Fraction.

4.2.5 Power input to the heating element.

4.2.6 Quality of Steam.

4.2.7 <u>Flow Measurement</u> :

For measuring water flow rate, a rotameter (0 to 2 gpm) was put in the wter line after the pump.

For steam flow measurement an orifice meter was designed and fabricated according to ASME specifications and then it was caliberated in the boiler house at different pressures by condensing and measuring the condensed steam. Boiler house condenser was used for the purpose.

4.2.2 <u>Temperature Measurement</u> :

Allthe temperatures of inner tube were measured with Cromel Alumel thermocouples of 22 gauge peened into the surface of the tube. For outer tube copper - constantan thermocouples of 24 gauge were used. The cold junctions of the thermocouples were put in cold - bath at 0° C., consisting of a wooden box filled with wooden dust and small ice blocks.

The reading of each thermocouple was taken by Milli - Volt-Potentiometer.

Four selector switches were used to readily pick - up the thermocouple at the desired location. The selector switches and the chart showing the numbers on selector switches corresponding the thermocouple locations on the test section surface are shown in the Figure 22.

4.2.3 <u>Pressure Measurement</u> (Figure 7)

The static pressures at rotameter, before the orifice meter, and right after the mixing section (at the first section of the test section) were measured with Bourdon gauges.

To measure the pressure drop between two sections 6" apart on the test section, three different arrangements were made (Figure 7). All the pressure taps were connected by copper tubing to a manifold through separate valves. When one valve is open the manifold feels the pressure corresponding to that particular point on the test section. To this manifold were connected a very sensitive Heise pressure gauge (accuracy ± 0.05 psia), a mercury manometer and a pressure transducer. Thus simultaneously three readings of the pressure could be taken. However, later the pressure transducer was not used because it started giving some trouble. It gave erratic readings for the pressure, due to some fault in the circuit fromthe transducer to the Oscilloscope (via charged amplifier). We did not find out the trouble because we were already having two reliable instruments for the purpose. Moreover, the accuracy obtained by pressure transducers for static pressure measurements was not acceptable(Accuracy was ± 2.5 psia). Pressure transducers are good for Oscillating pressure measurements (in transient states).

4.2.4 <u>Void Fraction Measurement</u> :

The equipment used for measuring the void fraction of the two - phase mixture is shown in block diagram of Figure 14. The radioactive (Gamma ray) source was Thulium 170, which was received in March 1970 from BARC with an original strength of 1 Curie. This source has a half life of 129 days and two energy peaks at 0.052 Mev

and 0.084 Mev. The gamma rays were collimated before entering the flow channel by a rectangular lead collimator whose dimensions and arrangement are shown in Figure 11. The source collimator and detector were mounted on a carriage (Figure 11) which could move in either a horizontal or vertical direction. The other accessories needed in the measurement of void fraction by this method are Gamma Ray Spectrometer, and Syntillation Crystal with Photomultiplier Tube (Figure [illegible]). Voltage stabliser was used for supplying power to this spectrometer at constant voltage.

4.2.5 Power Input to the Heating Element :

This was a 1 kw. electrical resistance type heating element 36 inches long providing a uniform heating along its length as discussed earlier. It needed a 110 V. supply, therefore a Variac transformer was used. Power delivered to the test section by this heating element was calculated from the measurements of voltage drop across the heating element and the current passing through it. A voltmeter and an ammeter were used for this purpose.

4.2.6 Measurement of the Quality of Steam :

Throttling Calorimeter was used to measure the quality of the steam entering the mixer and thereby the test section.

4.2 Experimental Procedure :

The boiler was fired, which takes about an hour to pick up pressures. In the mean time the water pump was run and the system (including mixer and test - channel) was bleeded off. Gamma - ray counts were taken (for half and hour) while the test channel was flowing full of water.

Getting the required pressure in the boiler, steam was allowed to bleed off the muddy condensates lying in the pipe line and then it was allowed to enter into the mixer and the test channel via cooling jacket. The quality of the entering steam was measured by throttling calorimeter. Flow rate and pressure of fog (wet - steam) in the test channel was controlled by two valves fitted at the entrance and exhaust sides of the test section. Proper care was taken to ensure that flow rate and pressure remained constant throughout a run. Time was allowed to stablize the flow by noting the fluctuations in pressure and temperature readings. After the system reached its steady state condition temperature, pressure drop and void fraction readings were taken.

The thermocouple cold junctions were put in ice - wells, and by frequently replacing the ice in the well, it was ensured that the junctions were keeping fully dipped in the ice (at 0 °C).

Pressure drop measurements were obtained by noting the readings of pressures at each of the six sections (on the test length) on both the Heise pressure gauge and the differential manometer.

For void - fraction, the counts of gamma ray (after it passed through the fog mixture in the channel) were taken for half an hour on the gamma ray spectrometer. Before taking the reading of gamma counts the spectrometer was put on for about half an hour to allow it to be stablised. Background counts and counts for the gamma beam after it passed through the channel filled with only air were taken frequently during the experimentation. Proper care was taken to ensure that the position of the gamma - ray beam passing through the test channel was fixed during a particular run.

CHAPTER V

RESULTS, CONCLUSIONS AND RECOMMENDATIONS

5.1 Results and Discussion:

The results of the present investigation have been presented mostly in the form of graphs and wherever necessary tables have been appended.

5.1.1 Pressure Drop Results:

Figures 5, 6A and 6B show the pressure drop data along the unheated annular tube for the two entrance pressures, 85 psia and 100 psia. The mass flow rates are 2.21×10^5 and 2.66×10^5 lbs/hr. ft.^{2} at 100 psia and 2.3×10^5 lbs/hr. ft.^{2} at 85 psia. The values of dryness fraction are 0.42, 0.51 and 0.59 at 100 psia and 0.416, 0.49 and 0.58 at 85 psia. The least square method has been used to correlate the data.

The results of Walli's homogeneous model (6), Owens homogeneous model (7) and Martinelli - Nelson Slip model (11) for pressure drop (for the same conditions as ours) have also been displayed in Table No. 1 alongwith the data of the present investigation to show the comparison. Levy's values of ϕ^2_{LO} (= $(dp/dz)_{TPF}$ /$(dp/dz)_{L}$ are too low and Bankoff's values of ϕ^2_{LO} are too high to make any sens

It is interesting to note that while the morphological structure of fog-flow seems to be closer to Homogeneous Model, the results of pressure drop are closer to the seperated flow model of Martinelli - Nelson. This can be explained on the assumption that in adiabatic fog flow at low mass velocities (lessthan the order of 10^6 lbs/hr ft^2) there exists a thin liquid film close to the walls of the channel thus providing more or less an annular flow pattern. This explanation is reinforced by the observations of Silvestri (3) and Collier (13). Silvestri (3) has reported that in adiabatic fog flow at mass velocities less than 10^6 lbs/hr. ft^2 there exists a thin liquid film having thickness of the order of 1 to 10 mils. Collier (13) has observed that homogeneous model correlations give better agreement in fog flow (over separated flow model correlation) at high mass-velocities (of the order of 10^6 lbs/hr. ft^2 and higher).

The results plotted on Figure 6A show the effect of mass velocities on the dimensionless pressure ϕ^2_{L0} for the fog - flow at 100 psia.

It is very interesting to note here that the dependence of ϕ^2_{L0} on the geometry (equivalent diameter of the channel D_e) and mass velocities at constant pressure and quality has been a matter of discussion and still is unresolved. While the analysis and correlation of Martinelli Nelson (11), Levy (4), Wallis (25) etc. shows no dependence of ϕ^2_{L0} on D_e and G, other workers like Marchaterre (31, 36), Mosettiola (reported from Wallis (25)) have

noted the dependence of ϕ^2_{LO} on mass - velocities. Marchaterre correlation (31, 35) has very strong dependence on mass-velocity which is not observed in the previous experiments. The present investigation supports the latter conclusion of dependence of ϕ^2_{LO} on mass velocities. For a change of mass velocity by a factor of 1.2 ϕ^2_{LO} also changes by a factor of about 1.2. This is in agreement with Muscettola observations (25).

Further, the experimental data have been arranged and plotted in Figure 6B to give more comprehensive idea about the pressure drops in fog flows. Pressure gradients have been plotted as a function of quality taking flow rates and pressures at entrance as parameters. The comparison of the values of pressure gradients at 100 psia and 25 psia for a given mass velocity (2.3×10^5 lbs/hr-ft^2) shows that the pressure gradient along the channel decreases as the pressure is increased. This is expected because the specific volume of steam (vapor-phase) increases at lower pressures causing a higher velocity of flow in the channel and thus causing higher pressure drop per unit length of the channel.

The plot of pressure - gradient versus quality at constant pressure (100 psia) (Figure 6B) taking flow rate as the parameter shows that pressure drop per unit length along the test channel increases with increasing mass velocity. This is obviously an expected result.

5.1.2 Void - Fraction Results:

The void - fraction measurements have been displayed in Figures 3 and 4. The percentage void - fraction at two pressures of 100 psia and 85 psia have been plotted as a function of percentage quality. The correlations of Owens homogeneous model (7), Levy (4) slip model and Martinelli - Nelson (11) separated flow model have been displayed graphically alongwith the data of this investigation for the sake of comparison.

As mentioned before, this investigation shows that the Martinelli - Nelson (11) separated flow model represents a closer picture of the actual phenomena in adiabatic fog - flow, at mass velocities less than about 10^6 lbs/hr.ft^2. The experimental results lie much nearer to the Martinelli - Nelson (11) correlation than Owen's homogeneous model correlation (7) and Levy (4) momentum - exchange model correlation. While Martinelli - Nelson correlation agrees with our results within about 1% difference, Levy's correlation shows about 3% difference and Owen's correlation shows about 4% difference. This discrepancy increases for these two latter correlations at lower qualities.

It is to be noted that the experimental data show a definite trend for the fog flow to lie in between the separated flow model and the homogeneous model.

Our data also shows (Figure 3) that, so long as the pressure remains constant (100 psia), the flow rate (mass velocity)

has no effect on the void - fraction. But this result may not be true if the flow rates are increased to high values (above 10^6 lbs/hr.ft^2), because then it is expected that the fog - flow resembles homogeneous model more than the separated model of Martinelli and Nelson (11).

5.1.8 <u>Temperature Distribution</u> :

Table No. 2 shows the measured temperatures on the outside surface of the inner tube and inside surface of the outer tube. The temperatures have been measured along the length of the test section at the intervals of 6 inches.

The actual plan was to collect data on heat transfer coefficients but due to unavailability of proper heating arrangement and breaking away of the tiny thermocouples fromthe inner tube during installations, we were forced to leave this idea for the present.

However, the result of temperature measurements, Table 2, shows the isothermality of the phenomena at steady state - which is expected. The difference in temperatures of the inner and the outer tube (at a particular cross - section) can be explained as follows:

The thermocouples were peened into grooves on the inner tube and therefore they measure the temperature at a certain depth in the wall of stainless steel tube and not at its surface. This

causes a drop in temperature due to thermal resistance of the tube wall material. Also, howsoever best is the fog, there is a possibility of a thin liquid film to occur on the surface of this tube and this phenomena further adds to the drop in temperature due to the thermal resistance of the liquid film.

On the otherhand, the bulbs of the thermocouples on the outer tube were protruding out from its inner surface, though thorough care was taken to fix them at the inner surface only. This protrusion will be enough for the bulbs to enter into the main stream of fog flow after penetrating through the thin liquid film formed on the inner wall of the tube. Thus these thermocouples will read the temperatures of the main stream of the fog, whereas the thermocouples on the inner tube will measure the temperature of this main stream minus the drops caused by the resistance of the liquid film and the thickness of the wall.

The difference in the two measurements that we are getting (about 12°C) sounds reasonable on theabove mentioned grounds.

5.2 Conclusions :

(i) On the basis of the literature survey on two - phase flow (and its particular case of fog - flow), it can be concluded that very little data are available on fog - flow and further that there is almost no data on fog - flow

below 100 psia and hence such data are needed for obtaining accurate and reliable design procedures for fog - flow systems.

(ii) The Martinelli and Nelson (11) correlation, (which was proposed on the basis of data on annular flows) gives the best agreement to the data of pressure - drop and void - fraction measurement in the range of variables of the present investigation. This conclusion, is once more, in support of the assertions of Collier (18), Tong (8) and Becker (19).

The pressure drop data are in agreement with Martinelli - Nelson (11) correlation within 25% (Experimental values are higher than the predicted values).

Previous to the present work, no evidence was available to ascertain that Martinelli - Nelson (11) correlation would satisfactorily predict the pressure drop and void-fraction in fog - flow at low pressures (below 100 psia) in annular geometry. Therefore, the goal of this research is believed to have been reached in as much as equipment was built where fog -flow in annular tube at 100 psia and below were obtained and it was established that Martinelli Nelson correlation makes satisfactory predictions of pressure drop and void fraction distribution.

(iii) It is also concluded that the non - dimensional pressure drop ϕ^2_{lo} $(= dp/dz)_{TPF} / (dp/dz)_{lo})$ depends on mass

velocity at a given pressure (100 psia) and qualities in the range of this experiment. For an increase in mass velocity by 20% ϕ^2_{LO} decreases by 30%.

(iv) At a given pressure (100 psia) void - fraction does not depend on the mass velocity; however it shows dependance on the pressure variation. As the pressure is decreased, the void - fraction increases for a given mass velocity.

(v) It can also be concluded that the pressure gradient (dp/dz) at a given flow rate is more at the lower pressures than at the higher pressures.

5.2 Recommendations and Scope for Future Work :

It is not very difficult to visualise the amount of experimental data that is lacking in the area of fog - flow and the urgent need to fill this gap. Although there are many aspects of fog flow which need investigation, a few pertinent problems that can be suggested for further investigation (and some of which could not be investigated in the present work due to the lack of time, complications during experimentation and restricted capability of our set up) are given below :

(i) To investigate the heat transfer characteristics of fog flow below the burn out conditions, at lower and higher pressures (lower pressure means at and below 100 psia; higher pressures mean above 500 psia). A very interesting

effect in regard to heat transfer coefficient in fog - flow below burn out condition has been reported by Sefar (8), (called "Hysteris Effect of heat transfer coefficient - this has been explained in the literature survey section). A study of this effect will help in understanding the basic mechanism of heat transfer in fog - flow phenomena.

(ii) The other area which needs to be explored is the prediction of burn - out conditions in low pressure regions. This could not be investigated in the present work due to experimental inadequacy.

(iii) The widely discussed problem of the dependence of the non-dimensional pressure coefficient ϕ^2_{LO} on the mass - velocities and geometry of the channel for a given pressure and quality of the fog - flow needs a very systematic investigation at both low and high pressures.

(iv) The most basic and perhaps the most important need of the two-phase (including fog - flows) studies is to develop a more rational "physical Model" of the flow. The present models - homogeneous, separated flow and variable density are either the one extreme or the other of the actual situation and are thus unrealistic.

It seems, to the author, that the effort of Bankoff (26) and Zuber et al (41) in resulting with a compromise

(variable density model) for fog - flow or bubble flow opens up a new approach. It may be possible to work out a new model essentially on the lines of Bankoff but by incorporating certain more logical assumptions like the existence of slip in the flow and by proposing a reliable mathematical equation for the radial variation of velocity and void fraction on the basis of experimental results.

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APPENDIX

GAMMA - RAY ATTENUATION TECHNIQUE
(Theory and Experiment)

A-1 Theory:

Gamma - ray attenuation technique is widely applied for measuring void fraction in two-phase gas-liquid flows.

As gamma rays pass through matter they are absorbed and the intensity decreases exponentially with distance. The extent of absorption is proportional to the intensity of radiation and to the thickness of the medium at a given point (14); as well as the material. This is described by the following equation :

$$dN = -\mu N . dX \qquad (A-1)$$

where

N is the intensity of radiation

μ is the linear absorption coefficient

dX is the thickness of the material.

For finite thickness X, this is integrated to give

$$\frac{N}{N_o} = e^{-\mu X} \qquad (A-2)$$

where

N_o is initial intensity

N is the final intensity.

In the derivation of the above equations it has been assumed that the radiation is monoenergetic, the gamma ray beam consists of narrow parallel rays and that the two phase mixture can be represented in layers of gas and liquid perpendicular to the incident radiation.

If the radiation passes through the channel when completely filled with gas the attenuation is

$$\frac{N_g}{N_o} = e^{-\mu_g X} \quad e^{-\mu_s X_s} \tag{A-5}$$

where

μ_g is the linear absorption coefficient for gas phase

μ_s is the linear absorption coefficient of the channel wall material.

X_s is the total thickness of the channel walls, and

X is the breadth of the channel.

A similar measurement at the same position with the channel completely filled with liquid gives

$$\frac{N_l}{N_o} = e^{-\mu_l X} \quad e^{-\mu_s X_s} \tag{A-6}$$

here the subscript l stands for the liquid.

Finally, the measurement obtained with the channel filled partially with gas and partially with vapor (two-phase mixture)

is given by

$$\frac{N_m}{N_o} = e^{-\mu_l X_l}\, e^{-\mu_g X_g}\, e^{-\mu_w X_w} \qquad \text{(A-5)}$$

Now by algebraic manupulations of the equations (A-3, A-4, A-5), an expression for the local void fraction ($\frac{X_g}{X}$) - fraction of the path of gamma ray beam through the gaseous phase in the flow channel (13, 18) - can be derived.

$$\frac{X_g}{X} = \alpha = \frac{\ln \left(N_m / N_l\right)}{\ln \left(N_g / N_l\right)} \qquad \text{(A-6)}$$

Thus, with the help of equation (A-6) it is possible to determine the local void fraction by making three radiation measurements to obtain N_g, N_l and N_m.

For getting the mean or overall void-fraction at a particular cross-section (for the entire cross-section), a series of count - rate measurements at different positions along the diameter or width of the channel should be made and then the average calculate

A-2 <u>Statistical Errors</u>:

Since the complete elimination of errors in physical measurements is impossible, it is essential to specify the probable error in reporting the result of measurements. The quantities frequently used in reporting statistical errors in measurements are described below:

(a) <u>Standard Error:</u>

For Poisson distribution the standard deviation, σ, is given by (17)

$$\sigma = (\bar{n})^{1/2} \qquad \text{(A-7)}$$

Where $\bar{n}$ is the average number of counts for the given time interval.

The physical significance attached to the standard deviation in a normal distribution is that 68.3 percent of all observations will fall within $\pm\sigma$ of the mean. Usually the true mean is not known and a single measurement of n counts is made. This value is reported as $n \pm n^{1/2}$.

Stated in terms of percentage, the percentage standard error = $\frac{100}{n^{1/2}}$ (A-8)

(b) <u>Probable Error</u> (18)

The Probable error is, by definition, exactly as likely to be exceeded as not. This error which has exactly a 50% chance of being exceeded in a normal distribution is equal to 0.6745.

$$\text{The Probable Error} = \frac{0.6745}{n^{1/2}} \qquad \text{(A-9)}$$

In our measurements, where the total number of counts is of the order of 2.2×10^6 (for 30 min.), the errors are -

Standard Percentage Error = 0.07

Probable Percentage Error = 0.046

Here the background count has been neglected in reporting the errors. They were of the order of 300 counts/min.

TABLE 1

VALUES OF ϕ^2_{LO} FOR STEAM WATER MIXTURES PREDICTED BY DIFFERENT MODELS (CORRELATIONS)
(For Fog - Flow)

Pressure 100 Psia

Steam Quality %	Wallis Homogeneous Model (25)	Owens Homogeneous Model (7)	Levy's Slip Model (4)	Bankoff's Model (26)	Martinelli Nelson Model (11)	Experimental values (at diff. Mass Velocities) 2.24×10^5 lbs/hr. ft^2	2.60×10^5 lbs/hr. ft^2	2.86×10^5 lbs/hr. ft^2
40.3	66.08	102.12	30.4	[illegible].7	117.0	–	–	137.0
46.0	67.94	104.87	36.7	1869.5	121.0	175.0	150.0	–
49.5	75.02	123.10	40.2	1770.0	142.0	–	–	155.0
51.0	79.05	128.94	46.4	1912.0	148.0	206.0	190.5	–
55.2	87.00	146.40	64.7	2404.0	169.0	–	–	185.0
58.0	88.52	148.32	67.4	[illegible].0	172.0	240.0	206.0	–

Pressure 85 Psia

Steam Quality %	Wallis Homogeneous Model (25)	Owens Homogeneous Model (7)	Levy's Slip Model (4)	Bankoff's Model (26)	Martinelli Nelson Model (11)	2.24×10^5	2.60×10^5	2.86×10^5
40.4	77.80	119.80	30.4	1678.0	136.0	–	–	160.0
41.3	78.90	132.80	40.3	1770.0	141.0	173.0*	–	–
46.0	88.85	144.47	48.0	2046.0	160.0	213.0*	–	–
49.2	96.35	148.48	58.4	2372.0	170.0	–	–	190.0
57.1	101.40	168.85	72.5	2688.0	195.0	–	–	227.0
58.0	102.45	172.82	78.5	3141.0	197.0	280.0*	–	–

* These values correspond to the mass velocity of 2.3 x 10^5 lbs./hr. ft. sq.

TABLE 2

VALUES OF TEMPERATURE (IN °C) MEASURED AT THE INNER AND OUTER TUBE OF THE ANNULAR CHANNEL
(ALONG ITS LENGTH AT DIFFERENT SECTIONS)

Length along the test section (in ft.)	Temperature at the outer surface of the inner tube (at different pressures and mass velocities)			Temperature at the inner surface of the outer tube (at different pressures and mass velocities)		
	100 Psia and 2.21x10^5 lbs./hr.sq.ft.	100 Psia and 2.00x10^5 lbs./hr.sq.ft.	85 Psia and 2.3x10^5 lbs./hr.sq.ft.	100 Psia and 2.21x10^5 lbs./hr.sq.ft.	100 Psia and 2.00x10^5 lbs./hr.sq.ft.	85 Psia and 2.3x10^5 lbs./hr.sq.ft.
0.0	x*	x	x	152.4	151.4	144.2
0.5	130.5	126.7	120.4	150.8	150.35	148.9
1.0	x	x	x	154.3	150.2	142.1
1.5	x	x	x	150.8	150.0	148.3
2.0	124.5	122.8	119.4	113.0**	103.0**	100.0**
2.5	125.1	124.0	118.3	148.9	150.1	148.7
3.0	128.2	128.0	117.9	150.05	149.6	148.0

* Thermocouples at these points were found not working after the installation of the test section.

** These thermocouples were showing inconsistent readings (may be due to improper contact with the surface).

BAKER'S PLOT FOR DISPERSED REGIO

100 PSIA

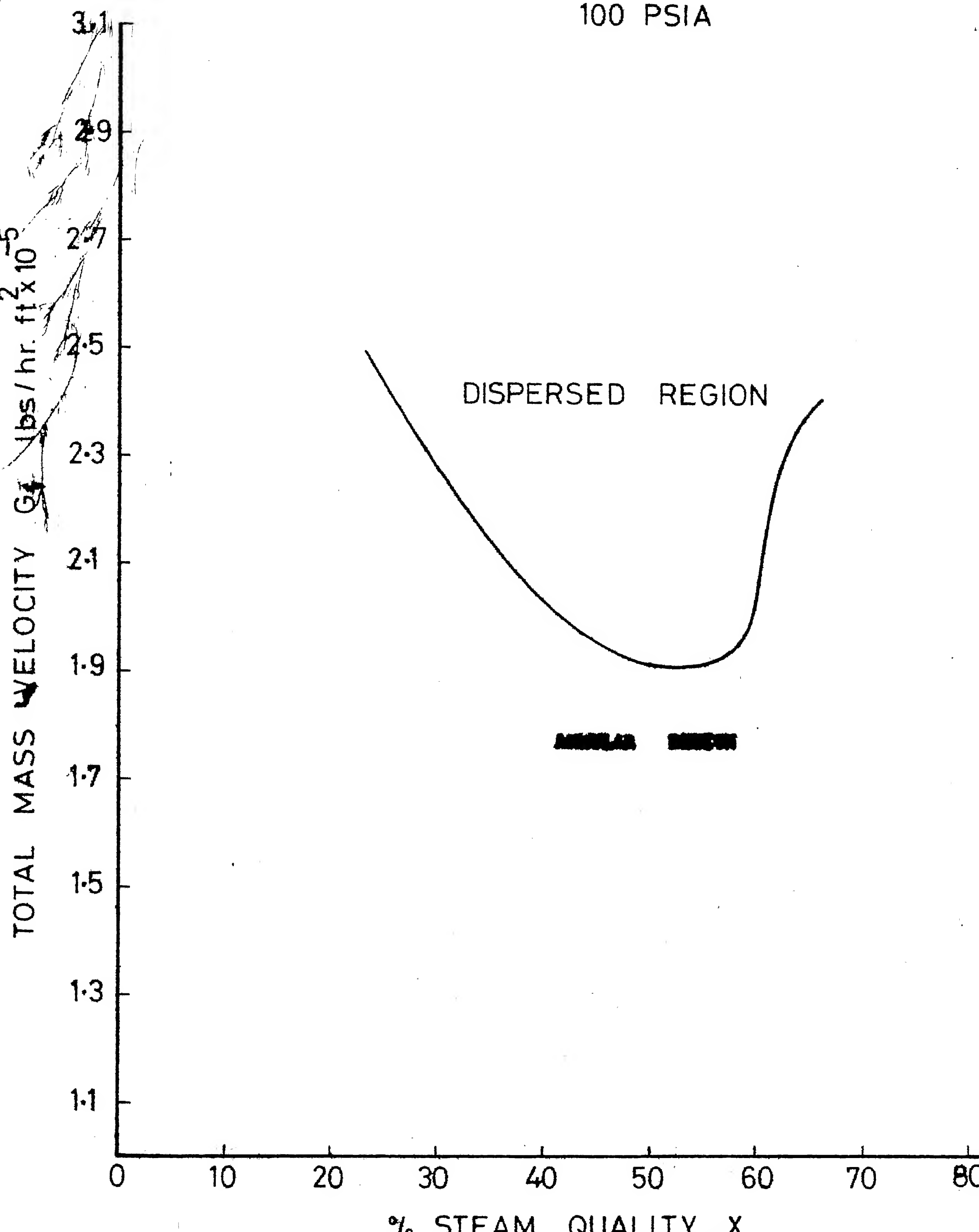


FIGURE 1

BAKER'S PLOT FOR DISPERSED REGION
85 PSIA

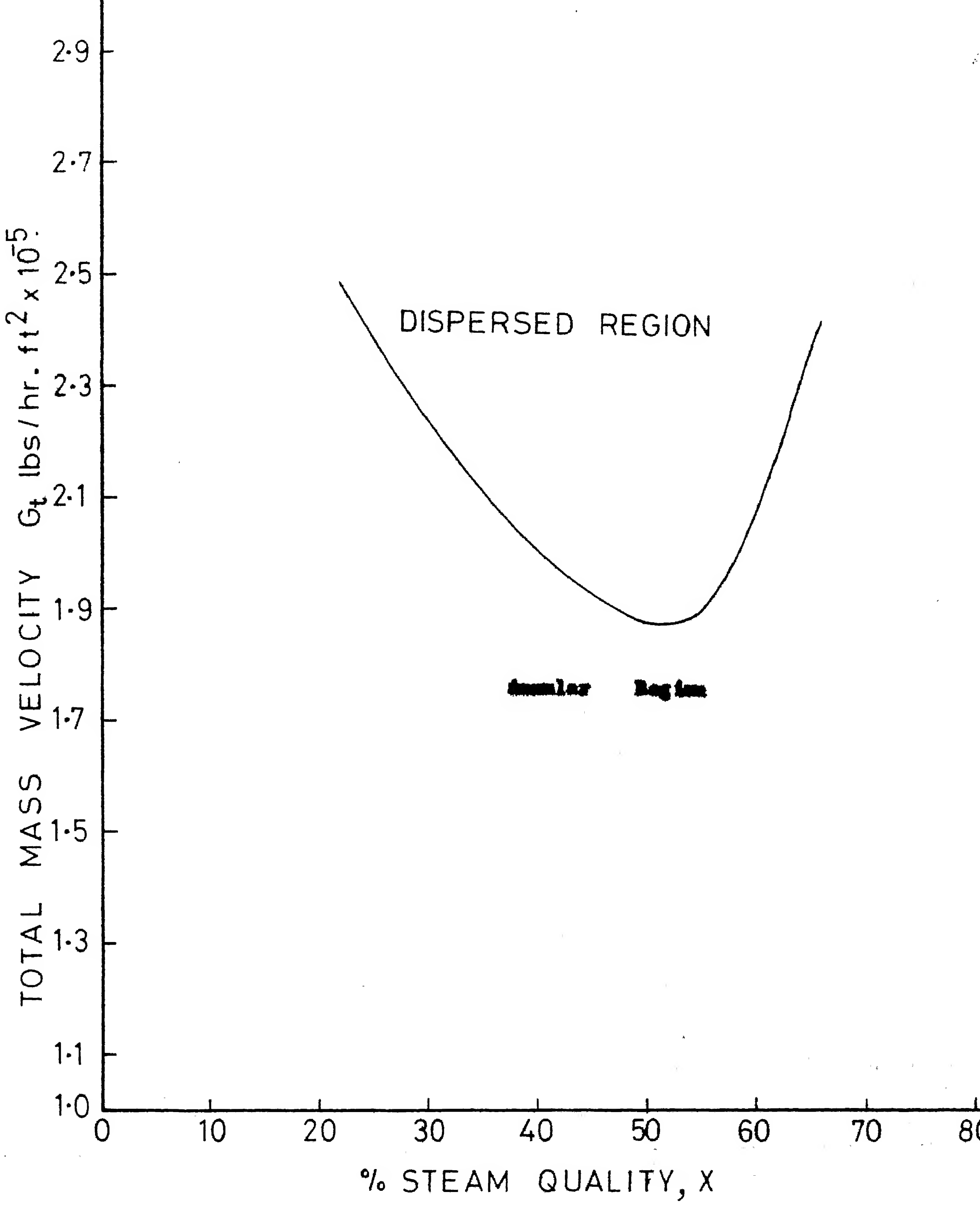
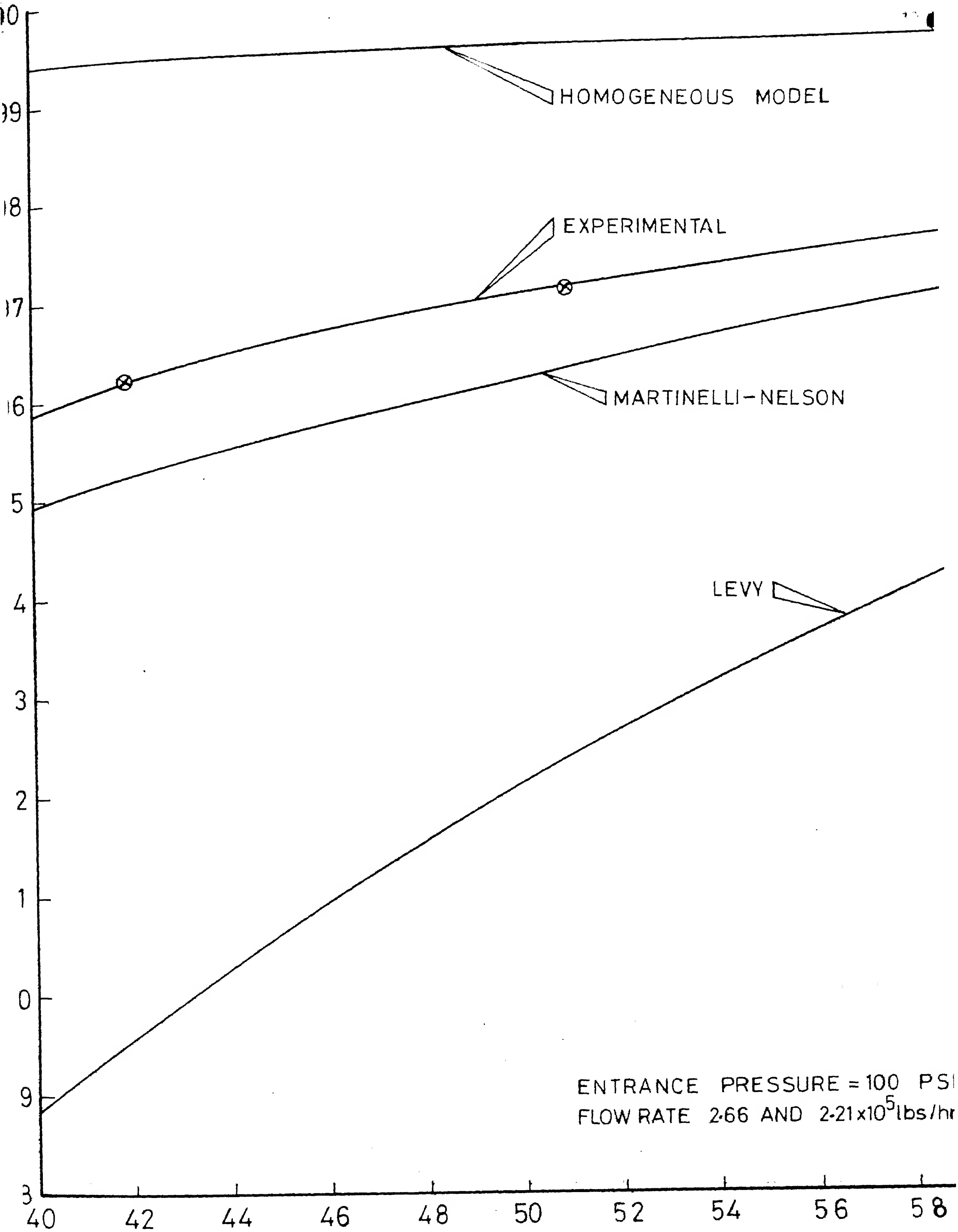
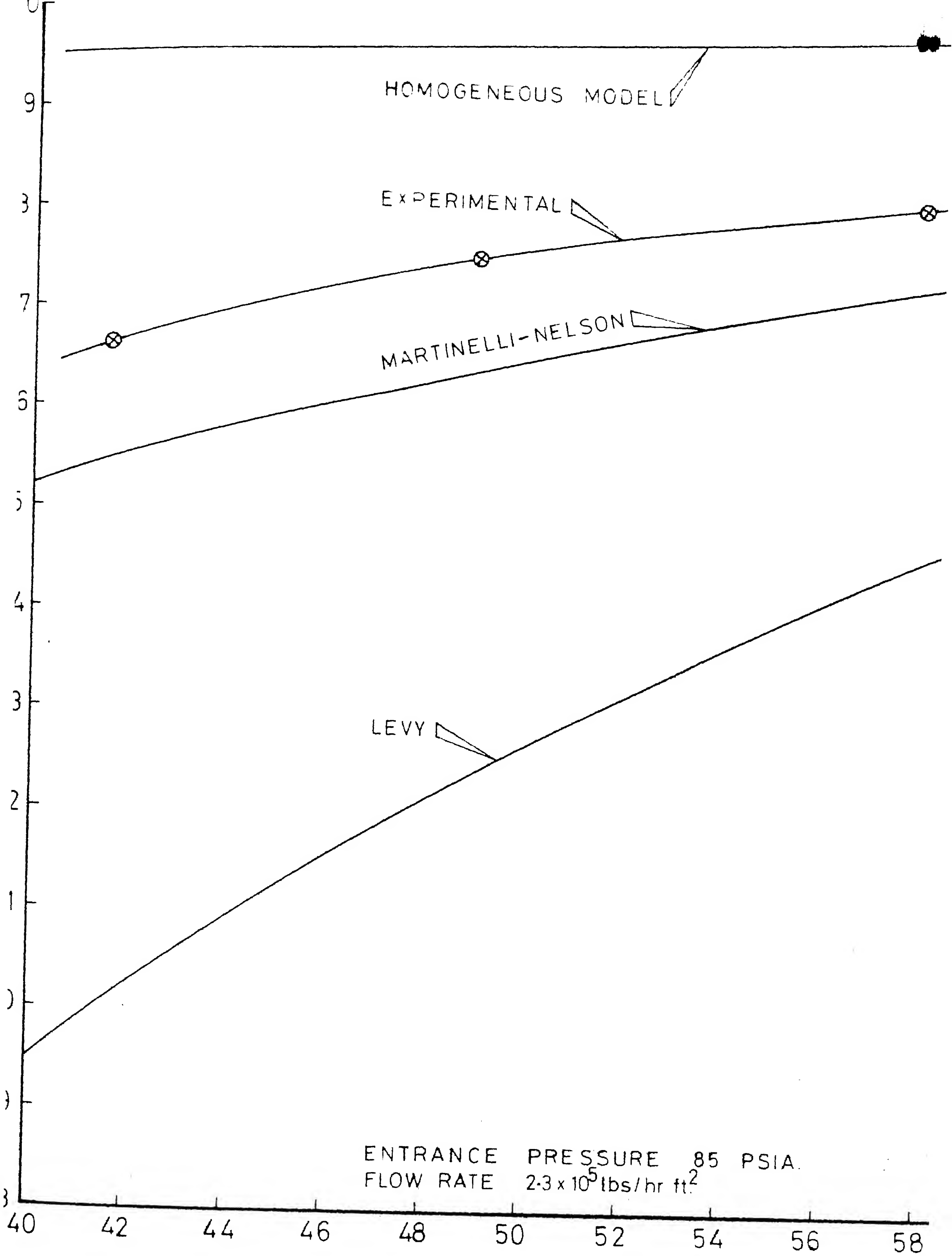


FIGURE 2



% STEAM QUALITY, X

VARIATION OF VOID WITH STEAM QUALITY



% STEAM QUALITY, X

VARIATION OF VOID WITH STEAM QUALITY

FIGURE 4

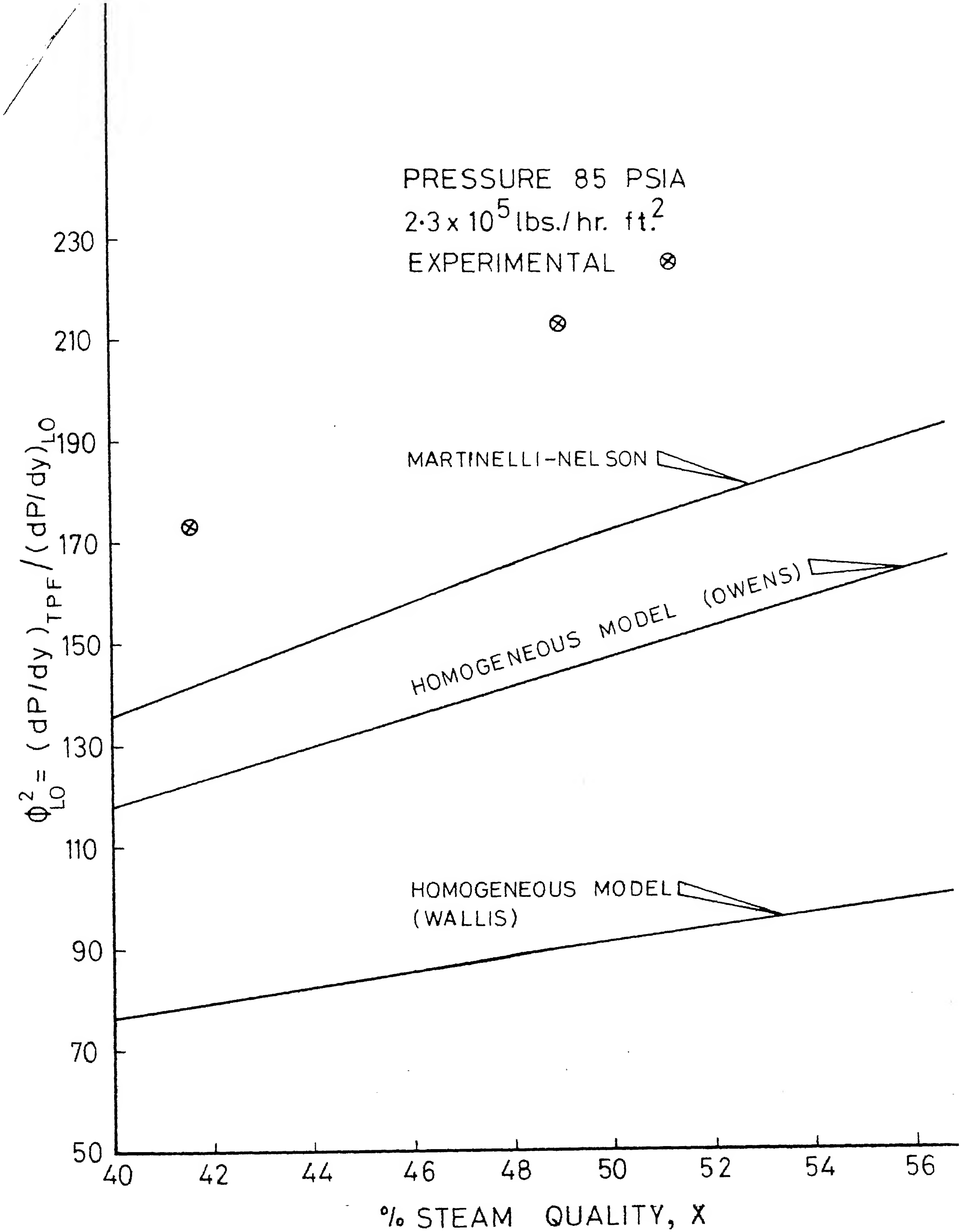
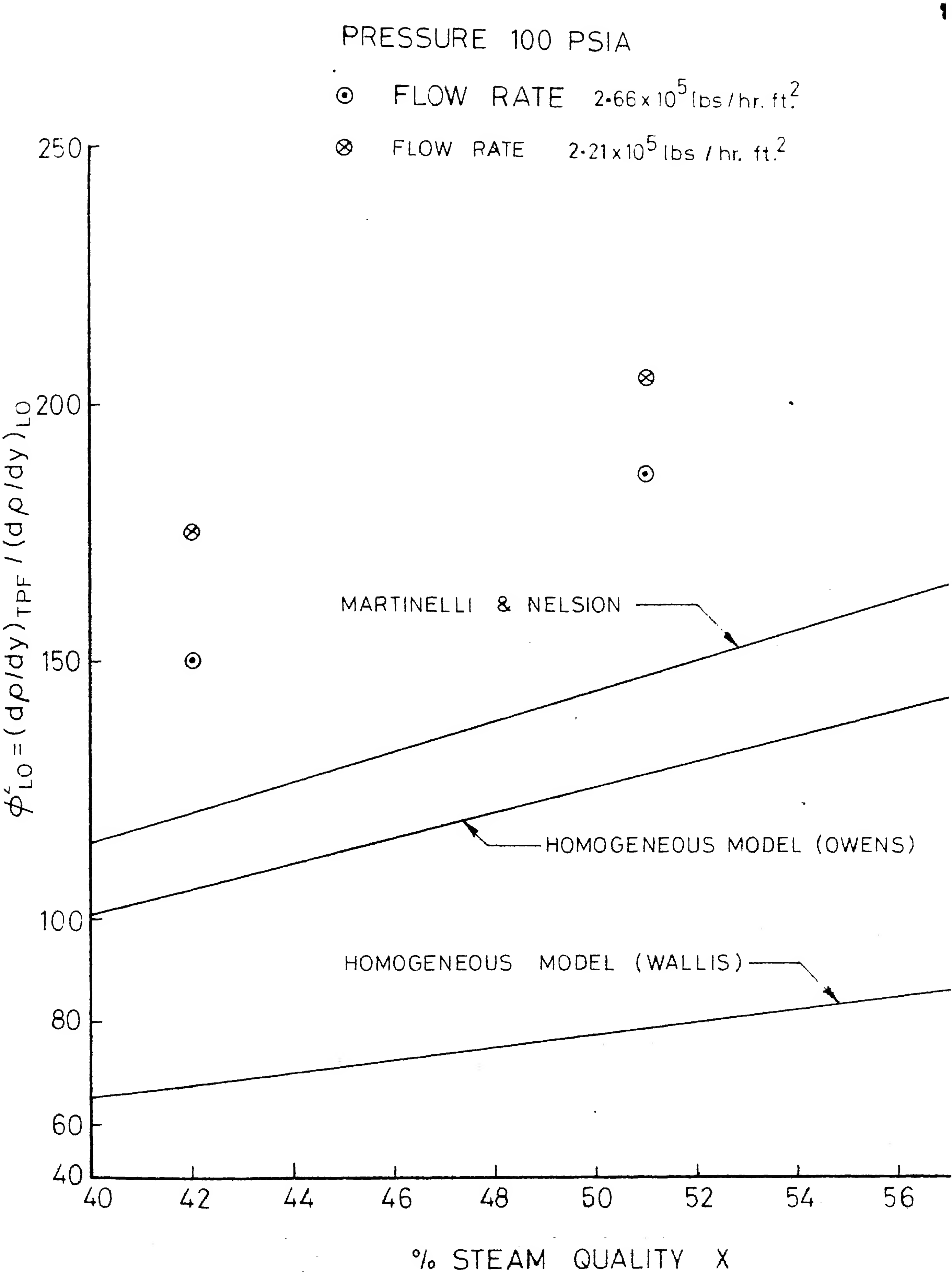
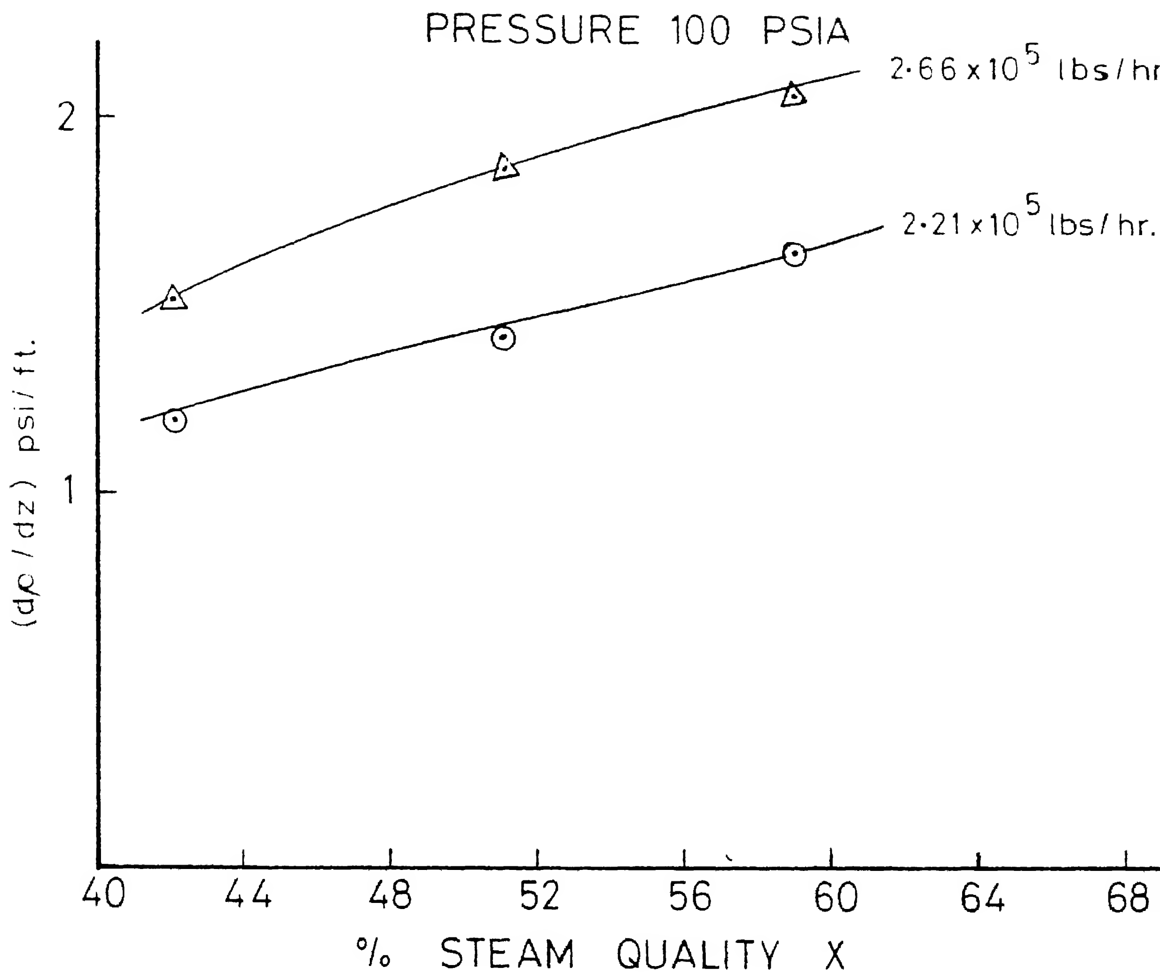

PRESSURE 85 PSIA
2·3 x 10^5 lbs./hr. ft.^2
EXPERIMENTAL ⊗
MARTINELLI-NELSON
HOMOGENEOUS MODEL (OWENS)
HOMOGENEOUS MODEL
(WALLIS)
$\phi^2_{LO} = (dP/dy)_{TPF} / (dP/dy)_{LO}$
50
70
90
110
130
150
170
190
210
230
40
42
44
46
48
50
52
54
56
% STEAM QUALITY, X

FIGURE 5



PRESSURE 100 PSIA
FLOW RATE $2{\cdot}66 \times 10^5$ lbs / hr. ft.^{2}
FLOW RATE $2{\cdot}21 \times 10^5$ lbs / hr. ft.^{2}
250
200
150
100
80
60
40
$\phi^2_{LO} = (d\rho/dy)_{TPF} / (d\rho/dy)_{LO}$
MARTINELLI & NELSION
HOMOGENEOUS MODEL (OWENS)
HOMOGENEOUS MODEL (WALLIS)
40
42
44
46
48
50
52
54
56
% STEAM QUALITY X

FIGURE 6 A


PRESSURE 100 PSIA
2.66×10^5 lbs/hr
2.21×10^5 lbs/hr.
(dp/dz) psi/ft.
% STEAM QUALITY X

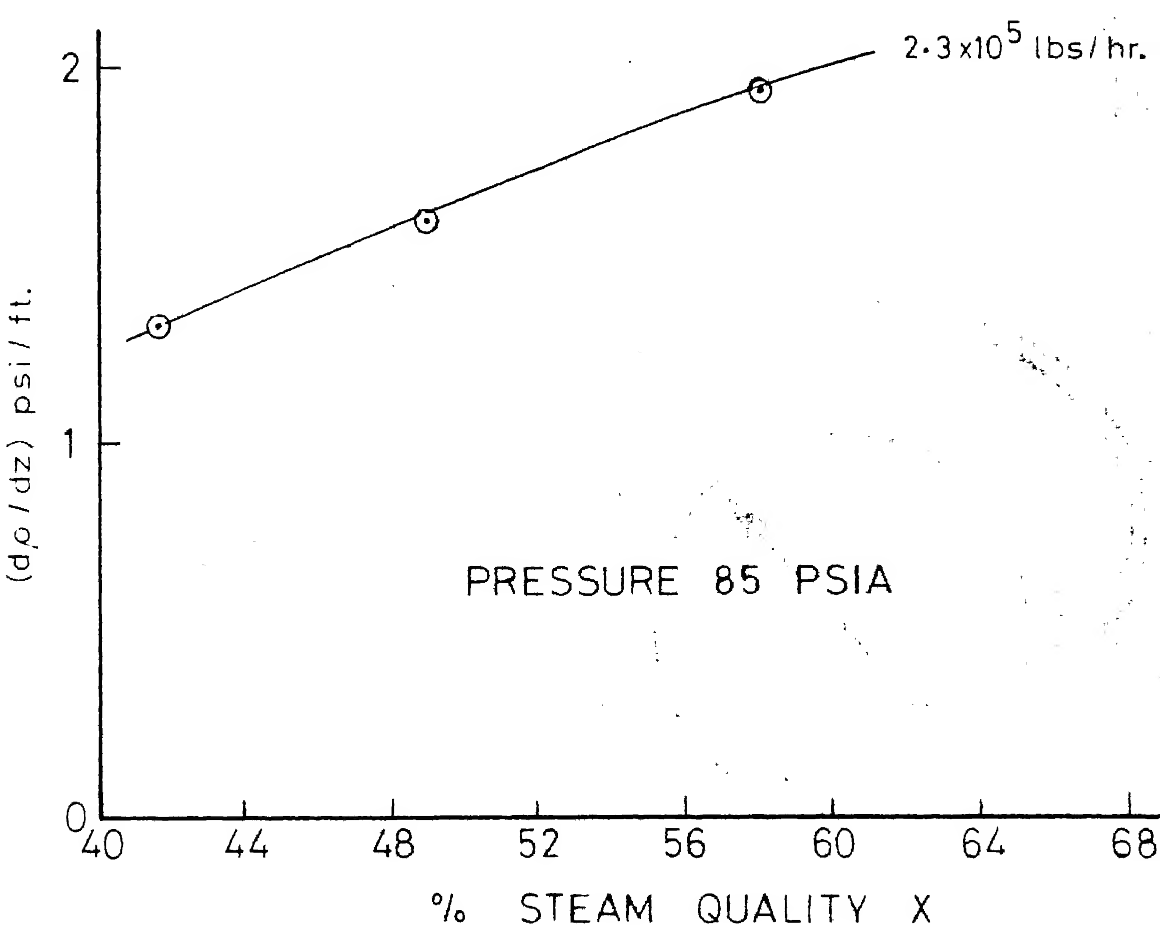
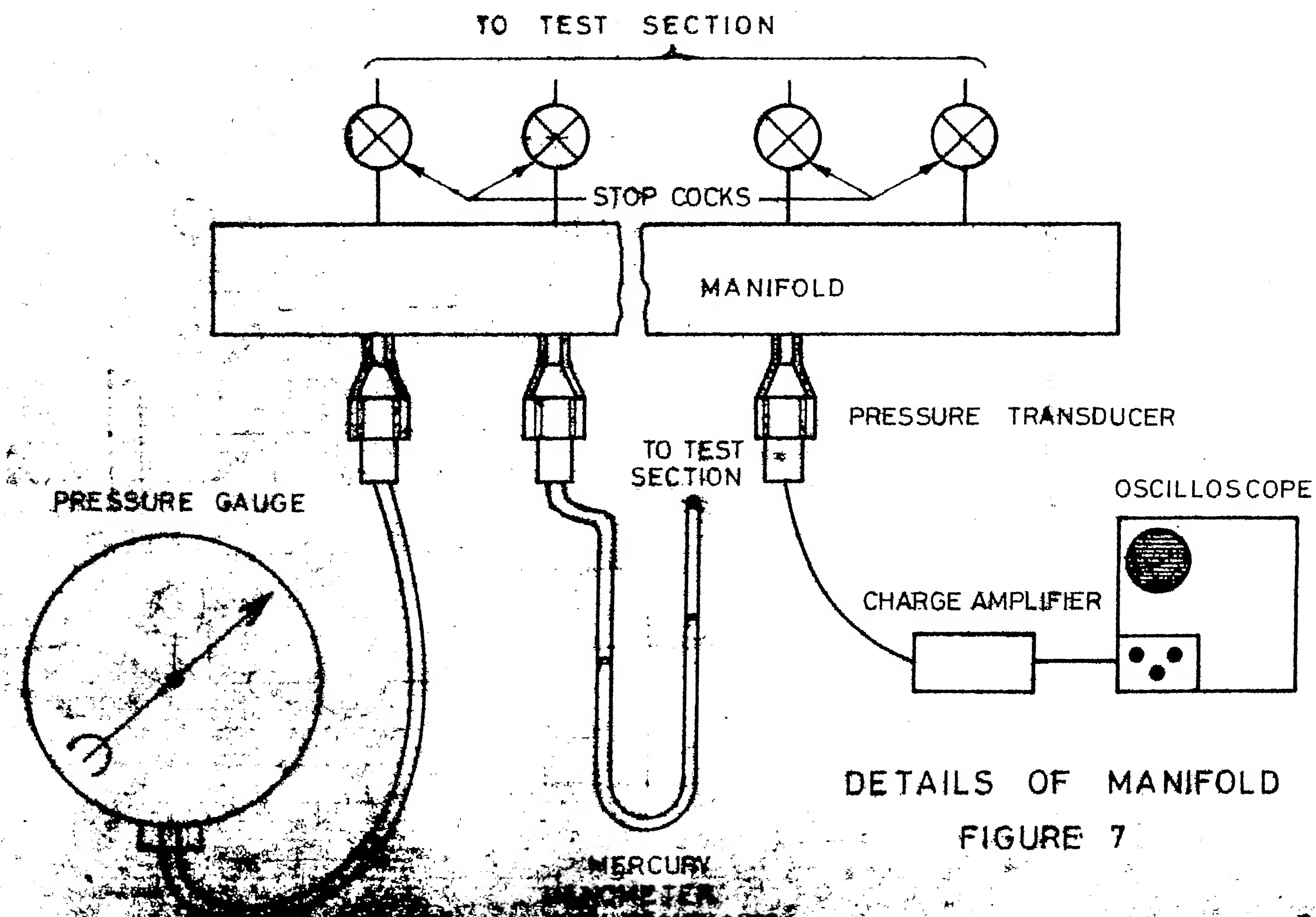
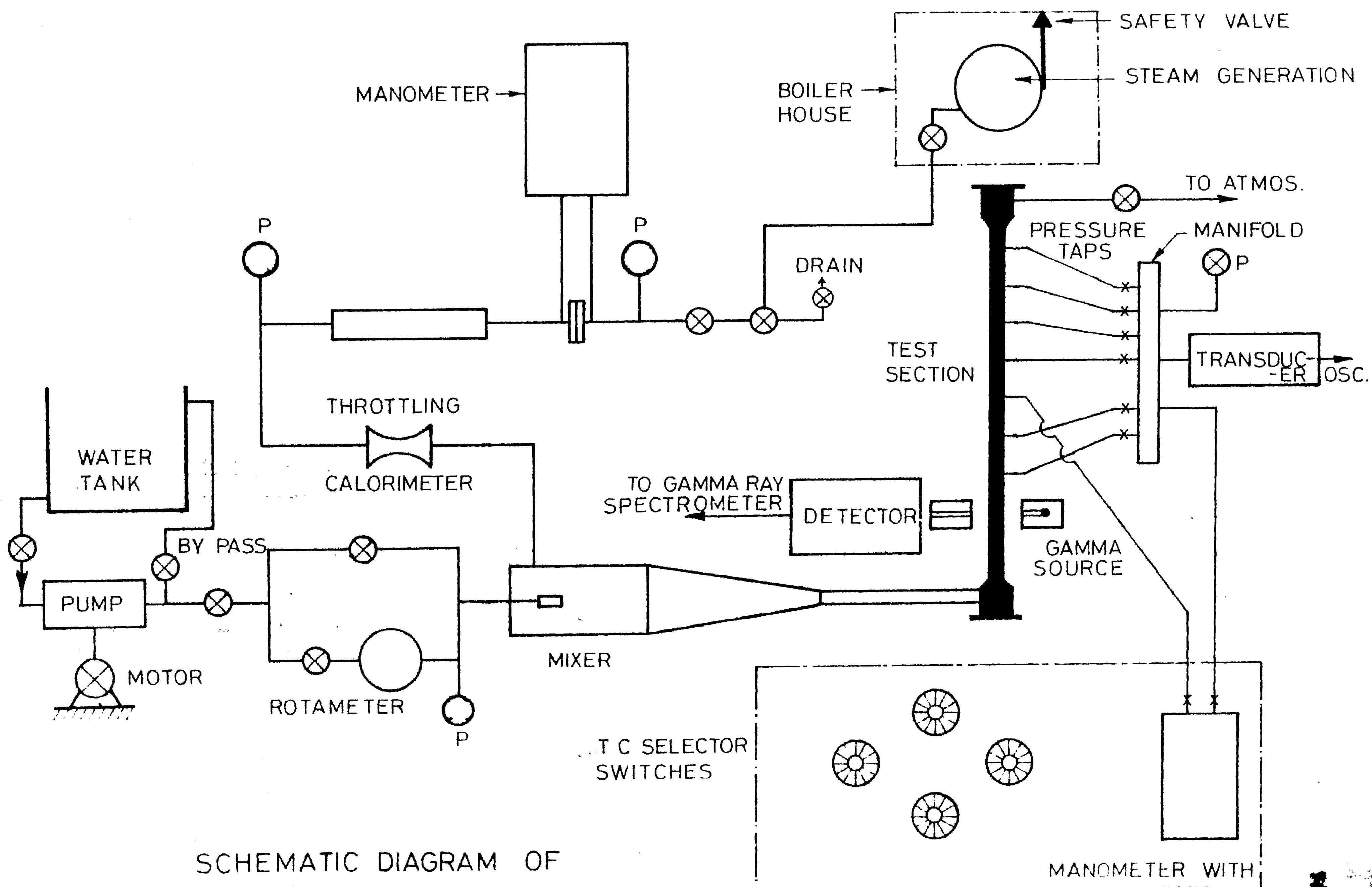

2.3×10^5 lbs/hr.
(dp/dz) psi/ft.
PRESSURE 85 PSIA
% STEAM QUALITY X

FIGURE 6-B

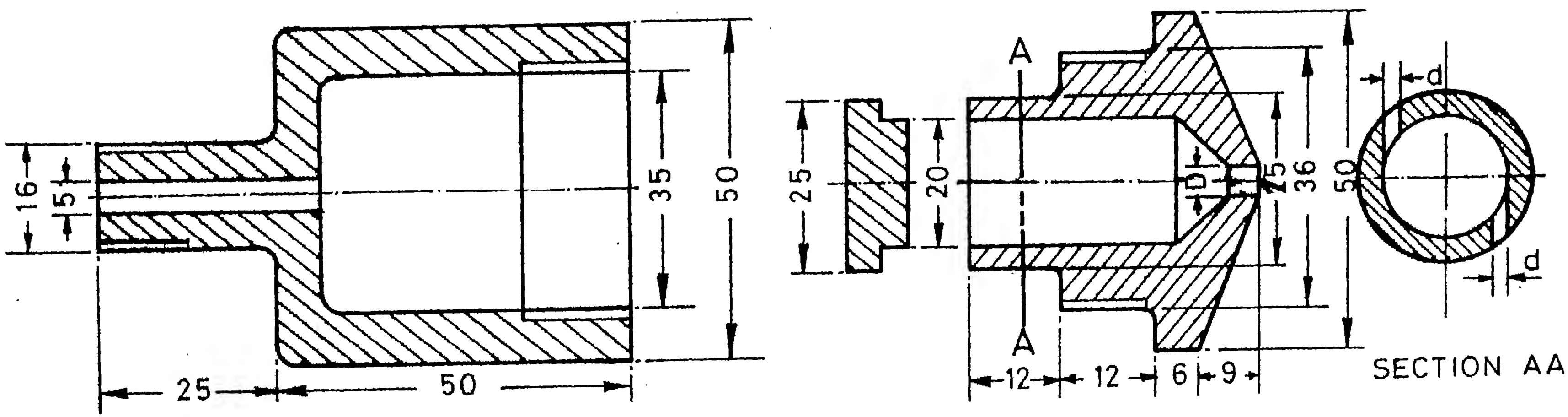


DETAILS OF MANIFOLD

FIGURE 7



SCHEMATIC DIAGRAM OF

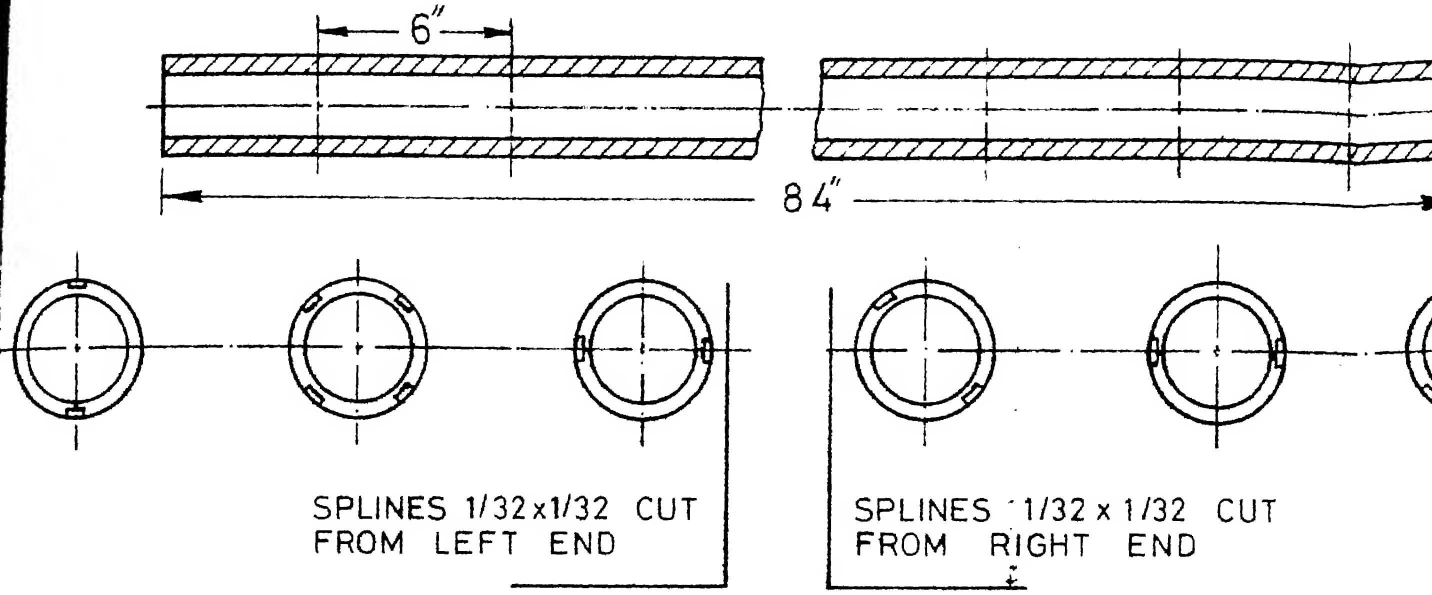


WATER SPRAYER

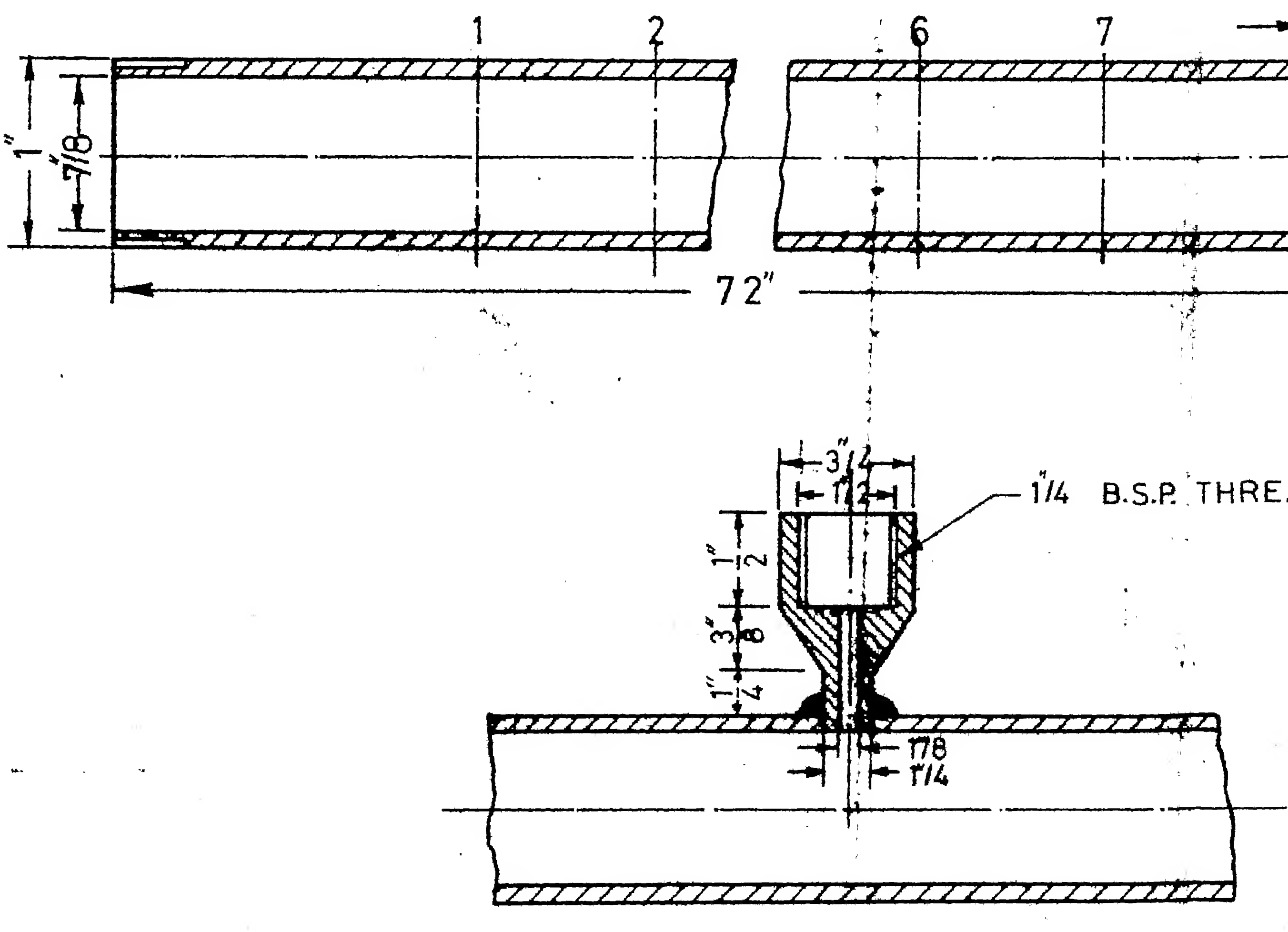
FIGURE 9

(ALL DIMENSIONS IN MMS.)

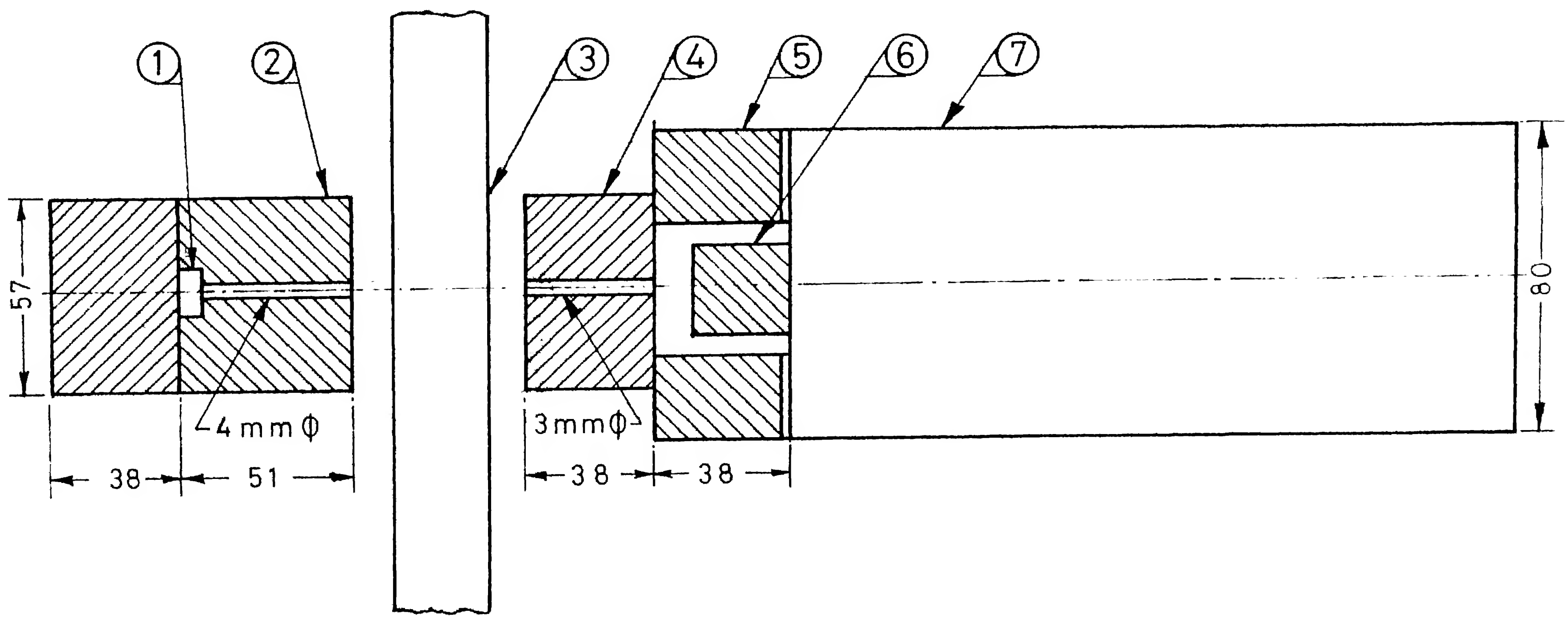
INNER TUBE
NO OF T C TUBES PEENED AT A SECTION
SHOWN BELOW



OUTER TUBE



D E T A I L S O F T E S T

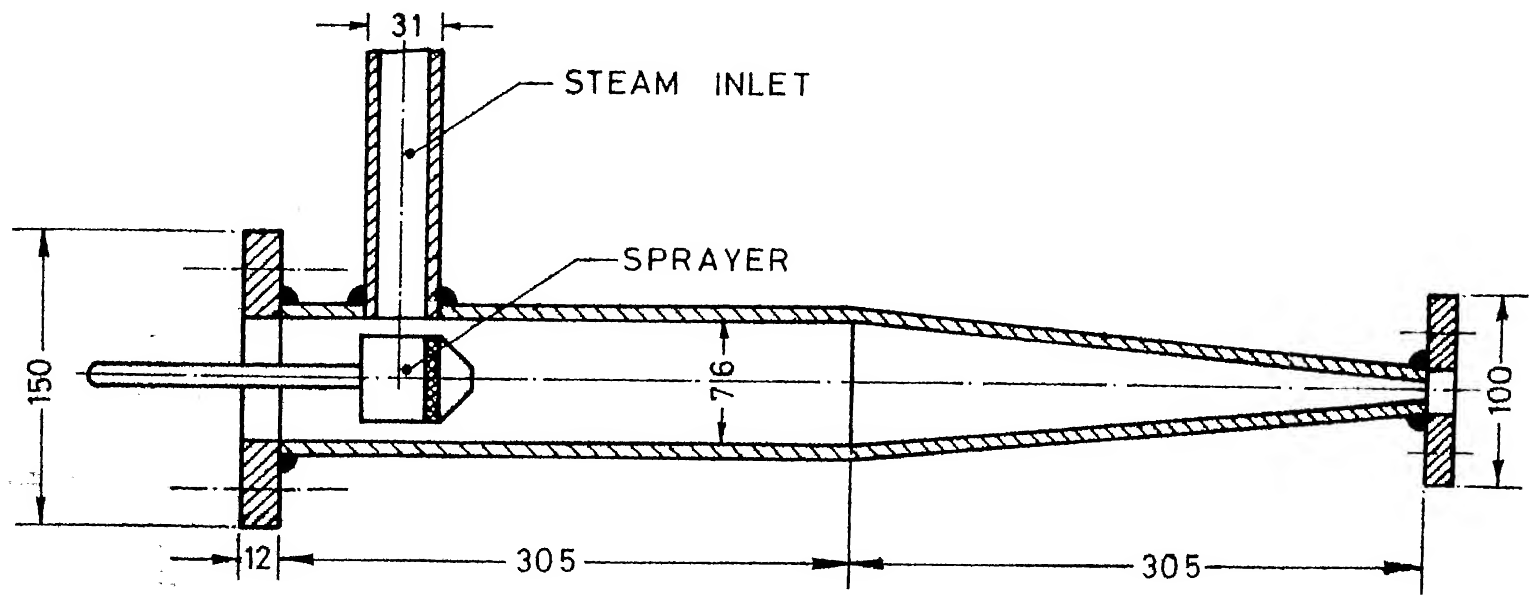


1 γ RAY SOURCE
2 COLLIMATOR
6 SCINTILLATION CRYSTAL

3 TEST SECTION
4 COLLIMATOR
7 PHOTOMULTIPLIER ASSEMBLY

5 LEAD SHIELD FOR SCINTILLATION CRYSTAL.

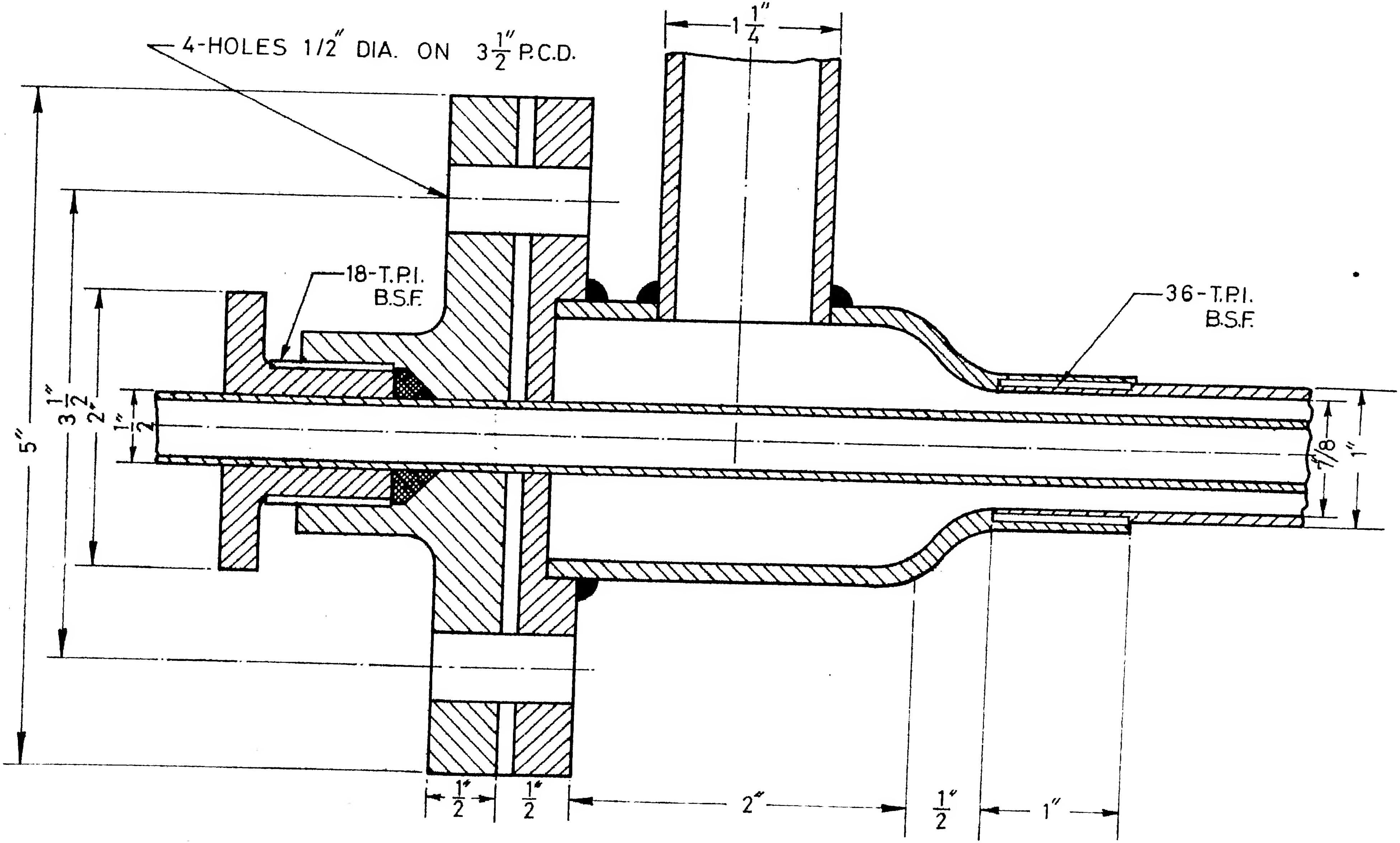
SECTIONAL VIEW OF THE SOURCE COLLIMATOR AND DETECTOR ASSEMBLY



MIXING CHAMBER

FIGURE 12

(ALL DIMENSIONS IN MMS)


4-HOLES 1/2" DIA. ON 3 1/2" P.C.D.
1 1/4"
18-T.P.I. B.S.F.
36-T.P.I. B.S.F.
5"
3 1/2"
2"
1/2"
7/8"
1"
1/2"
1/2"
2"
1/2"
1"

VOID METER EQUIPMENT
SCHEMATIC DIAGRAM

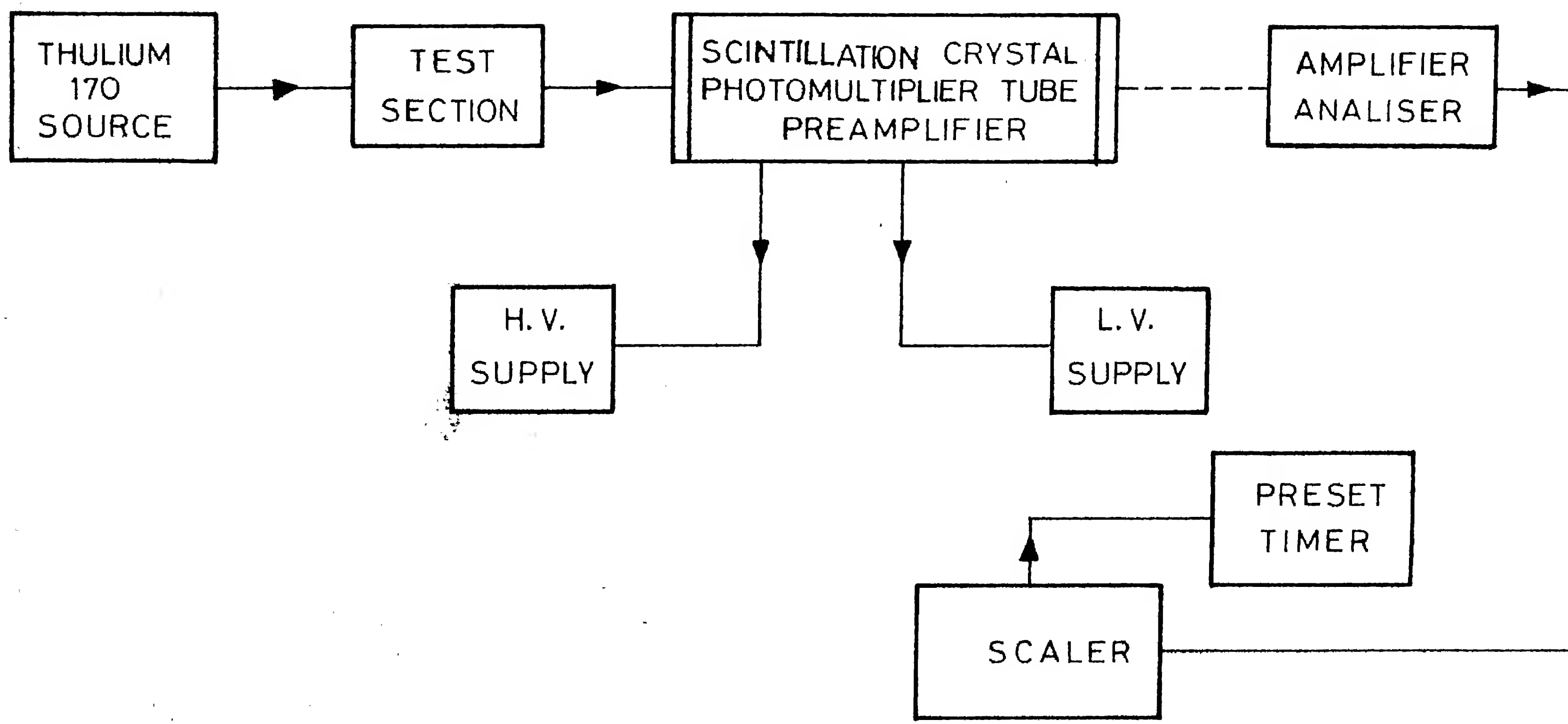
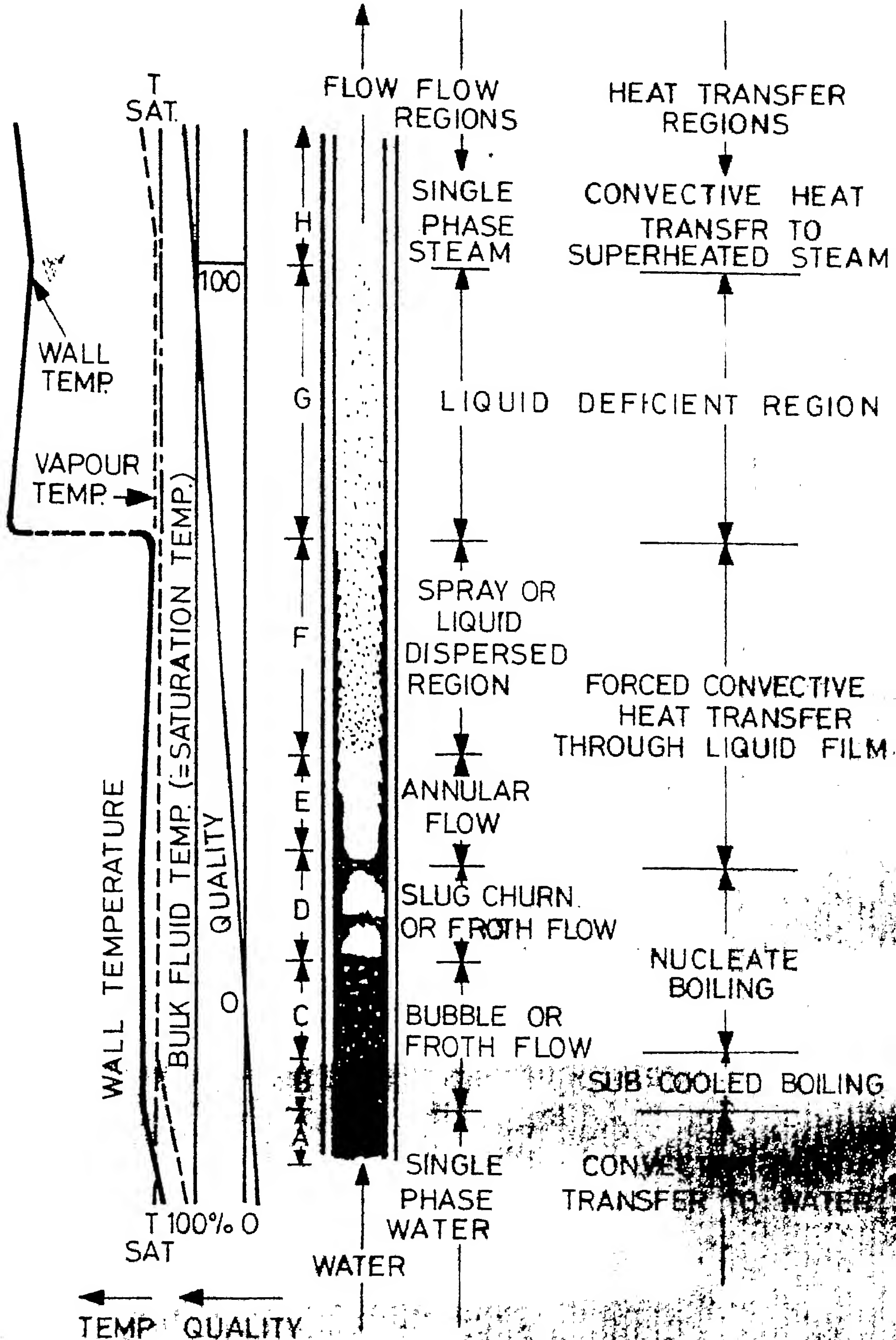


FIGURE NO. 14



REGIMES OF TWO PHASE FLOW

FIGURE 15

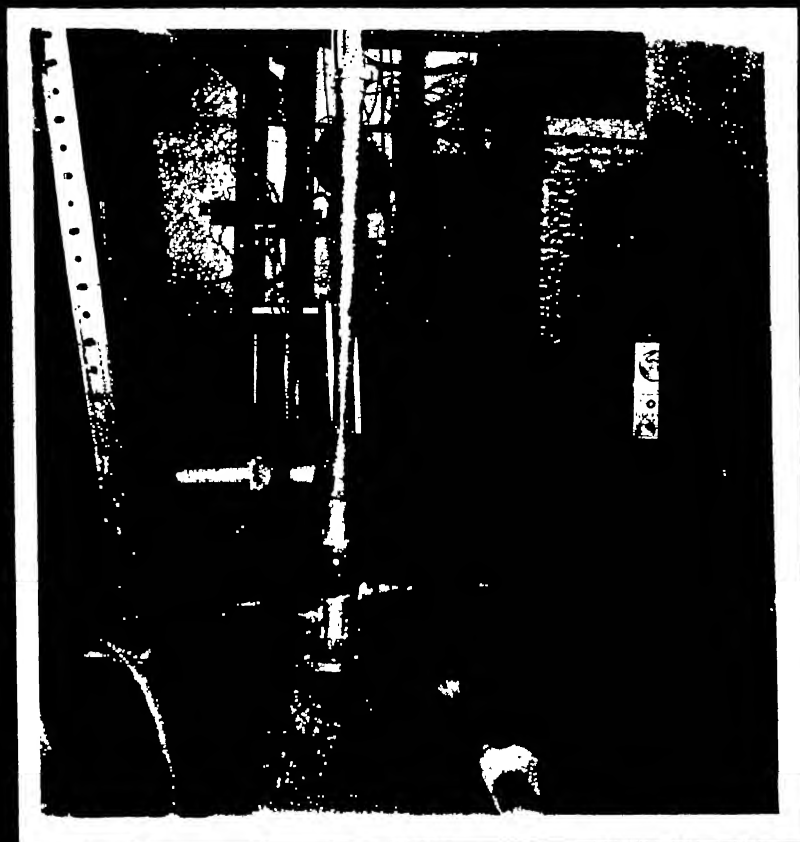


FIGURE 10

Full View of the [illegible]

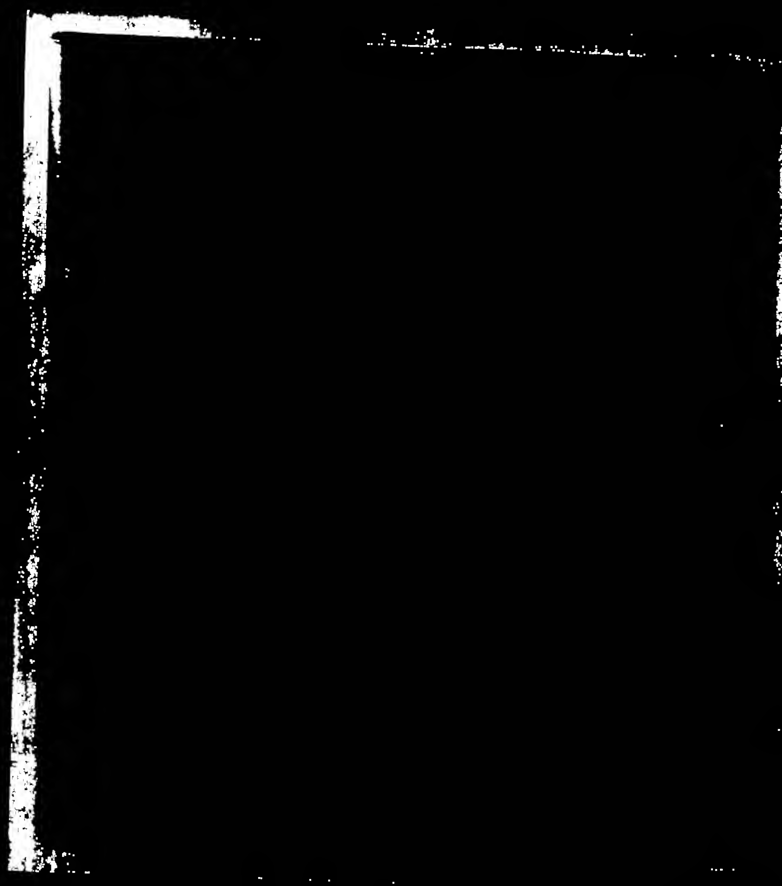


FIGURE 17

View of Centrifugal Pump and Motor [illegible]

FIGURE [illegible]

View of the Mixing Chamber

FIGURE [illegible]

View of Void Fraction Carriage Assembly and Test Section

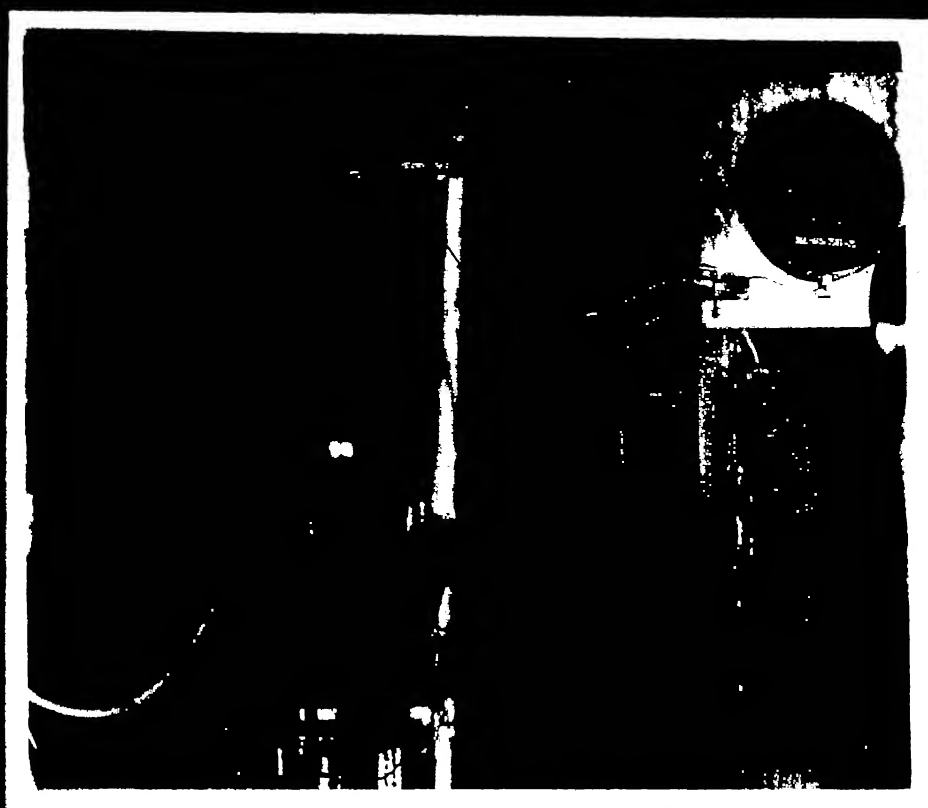


FIGURE 10

View of the Test Section and Instrumentation

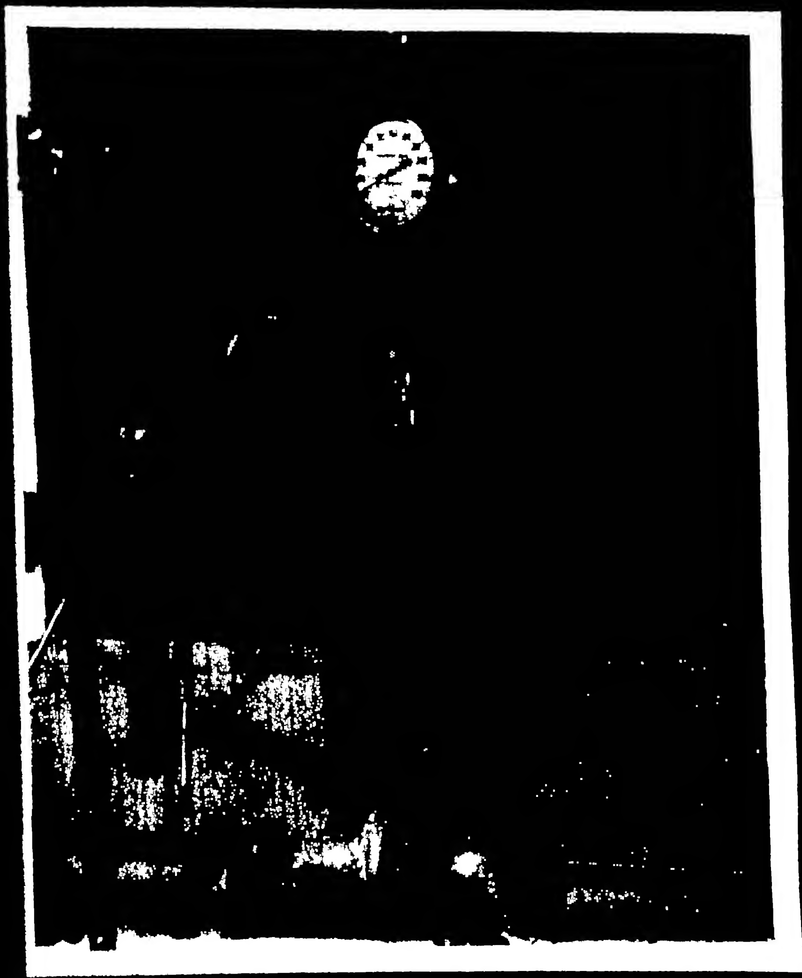


FIGURE 21

View of the Orifice Meter

FIGURE [illegible]

View of the Panel Board and Instruments.

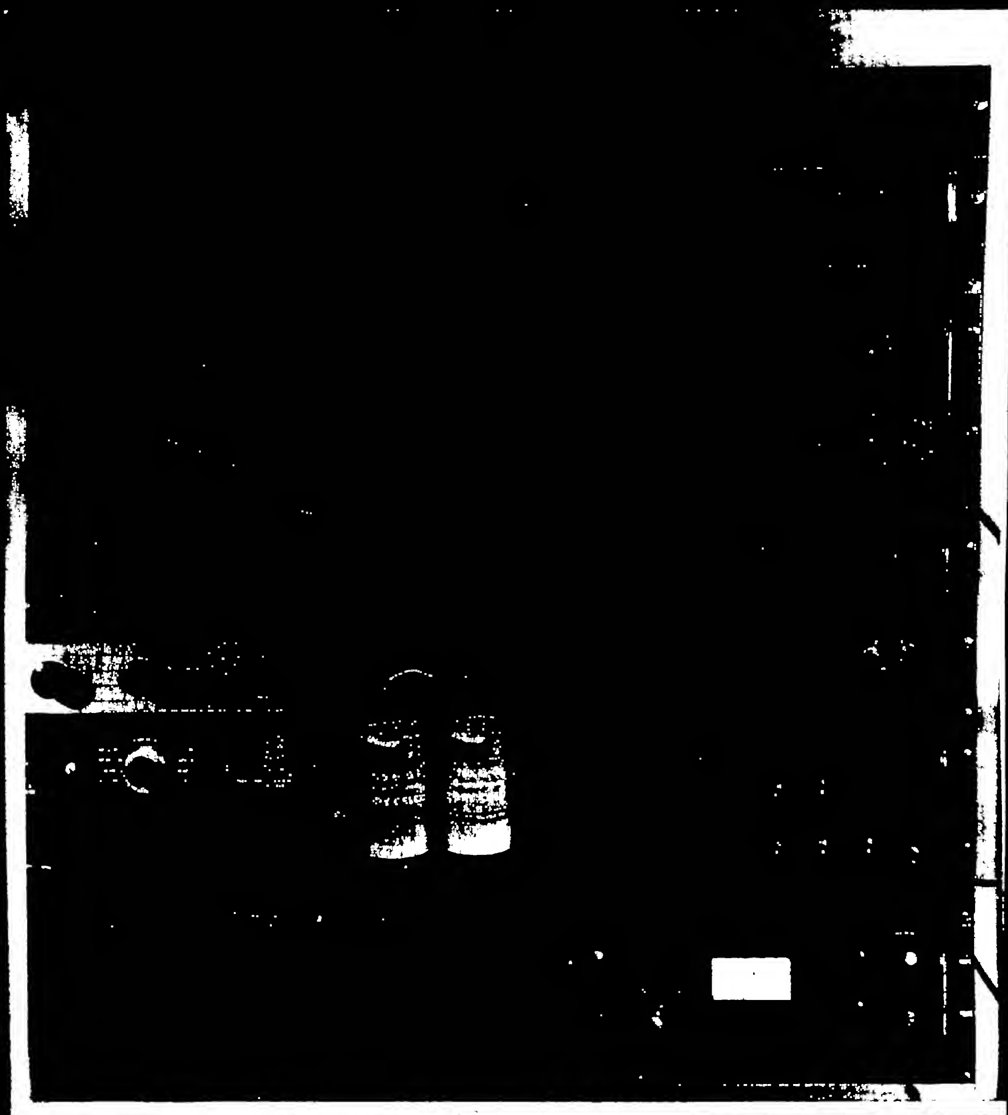


FIGURE [illegible]

View of the Instruments (Gamma - Ray Apparatus)

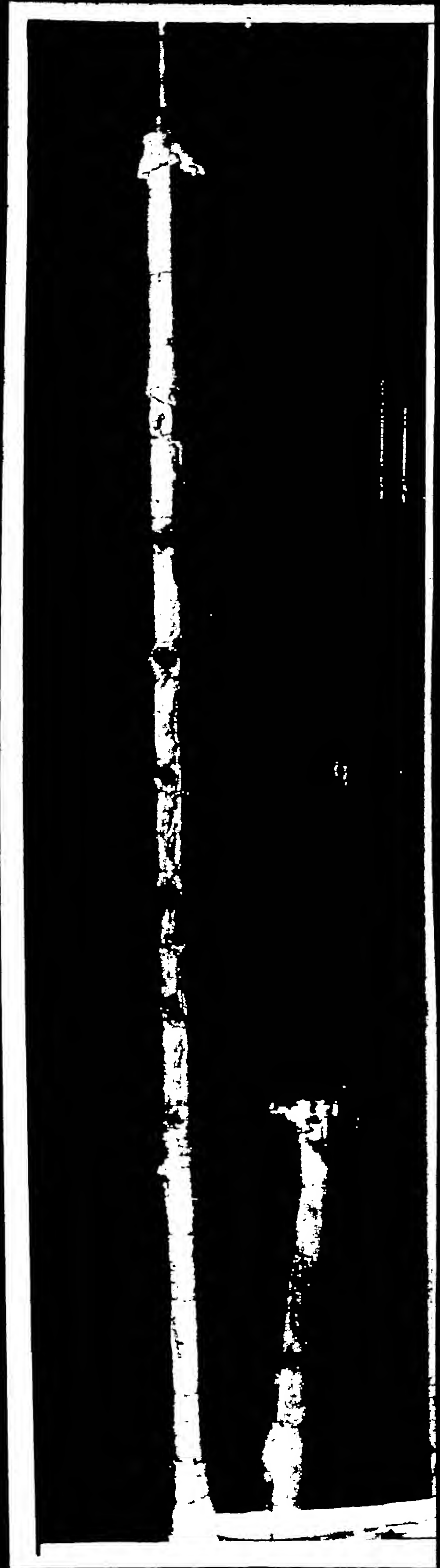


FIGURE [illegible]

View of the Test Section before installation